

DESIGN AND CONSTRUCTION OF AN  
INSTRUMENT TO DYNAMICALLY EVALUATE  
A FORCE PLATFORM

by

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## INTRODUCTION

Sophistication in the design of instruments has been the theme of the Space Age. This theme applies aptly to the force platform. Since its conception by Cunningham at Berkeley (Cunningham and Brown, 1952), refinement of the force platform has occupied the thoughts of several designers and researchers. The most recent design is by Hearn (1966).

Cunningham rightly should be called the father of the force platform even though today's design retains few of the initial design's characteristics. Some alterations of the original design have been: (1) force detection by measuring beam deflection with a linear variable differential transformer (LVDT), (Greene, 1957), instead of directly applying pressure to piezoelectric crystals (Lauru, 1957), (2) increasing the upper plate platform area from a triangular to an equilateral hexagon design to preload the vertical axis in both directions (Barany, 1962), (3) addition of torque measurement capabilities (Hearn, 1966), and (4) lead screw adjustment of LVDT (Hearn, 1966).

Even though the development of the force platform has had a normal technical growth and human engineers have utilized this instrument to study the work situations, there appeared to be a lag in developing and interpreting the constants of calibration to be used in calculating a force exerted by the subject. Intuitive reasoning indicates that a basic procedure such as calibration must be firmly established before conclusive results can be obtained from the force platform.

The primary purpose of this research was to evaluate the response characteristics of Hearn's platform through dynamic calibration. Comparison was made between dynamic and static calibration in order to investigate whether use of static calibration in the past was valid. Evaluation of dynamic calibration was in the form of limits that can safely be imposed upon

the instrument. The evaluation required:

1. Design of a dynamic calibration device
2. Construction of the calibration device
3. Evaluation of the calibration capability of the device for the platform

## STATEMENT OF THE PROBLEM

Hudson (1962), at the State University of Iowa, visualized the subject-platform system as being elastic, and thus potentially dynamic. His reasoning stemmed from the fact that the platform is supported by cantilever beams which are sprung or elastically stressed. The initial stress is produced when the weight of the upper plate is placed upon the beams, and more stress is added later during actual operation when the weight of a subject is added to the weight of the plate.

Hudson posed two questions as to the authenticity of the dynamic responses obtained from the structure: (1) To what extent does inertia effect the performance of the platform? (2) Does harmonic addition of displacement of the structure when dynamically loaded affect the operation of the instrument?

Hudson's approach to solution of the two questions was to theoretically determine the response characteristics of the apparatus by mechanics, and then use experimental evidence to substantiate his findings. The inertial effects encountered when working with the platform were almost impossible to determine theoretically. A theoretical solution was more nearly approached to the problem of harmonic displacement through the use of vibration mechanics. More specifically, a solution was obtained by applying the equations of natural frequency and dynamic amplification to the subject-platform system. Hudson found, from these equations, that there will be some amplification of displacement at any frequency, but only at 500 cycles per minute will the platform attain its natural frequency.

Phase two of Hudson's analysis was to substantiate his theoretical predictions through experimentation. This aspect of his analysis was done by Ivan Jacobson (1960).

Jacobson used a motor coupled to a variable speed reducer to dynamically evaluate a platform similar to the original one constructed by Greene (1957). The speed reducer was coupled to a slotted U-shaped yoke which could be adjusted to vary the stroke of a reciprocating input shaft. Next, Jacobson connected a calibrated spring between the input shaft and the force platform. Then, by varying the adjustment on the yoke and measuring the length of stroke for each adjustment, Jacobson determined the force input to the platform.

In summarizing his research, Jacobson felt that a frequency of force application should range between 50 and 100 cycles per minute (CPM). Errors in his recording equipment apparently caused doubt about the range from 0 to 50 CPM. In comparing static to dynamic calibration curves, Jacobson reported that dynamic calibration curves were as much as fifty per cent less than curves obtained statically with equivalent force applications. For example, a static force of 50 pounds would deflect the recording pen 10 millimeters, whereas a dynamic force application of 50 pounds would only deflect the pen 5 millimeters from a zero position. Jacobson also reported sensing a force of .5 pound in the lateral and frontal axes. The vertical axis was able to sense only a 1 pound force. In the summary, Hudson (1962) cites five main findings of the experiment. First, as stated by Jacobson, there appears to be an oversensing of forces at frequencies above 150 CPM. This effect was more pronounced when the platform was loaded with a 200 pound deadweight. Hudson's analytical model indicates that oversensing will occur at the higher frequencies. Secondly, Hudson states that some combination of inertial and frictional effects is attributed with causing a negative sensing to occur in force detection. Thirdly, peculiar to the platform design, there was sensing of forces in the unloaded axes during a period in which the loaded axes were oversensing a force. Again, as in the second finding, there was no conclusive

reason for this happening. Fourth, static calibration was distinctly different from dynamic calibration when finding a calibration constant. This fact became apparent after a plot of force applied (pounds) versus units sensed was made for static and dynamic calibration. Lastly, the difference between static and dynamic curves tends to increase as the forces applied increased. Explanation of this widening differential followed from the fact that the force-frequency device was an elastic system which adversely affected the constant obtained from the calibrated spring. This change in the spring constant was considered more of an inconvenience than a limitation.

Due to the discrepancy between static and dynamic force calibration, Hudson felt that future users of the platform should use dynamic calibration. If for no other reason than to evaluate the limitations of the force platform, the researcher should utilize dynamic calibration.

The most recent experiment which utilized both static and dynamic calibration as preliminary to actual experimentation was done at Arizona State University by Adams (1966). Although Adams fails to explain his reasoning for using either static or dynamic calibration, the mere fact that he uses both forms attests to the skepticism for using only static calibration.

Adams used static calibration to establish a constant by which a later force can be determined. In his application, dynamic calibration was used only to determine whether the frequency at which his subjects would be performing approached the natural frequency of the force-detecting unit. Although his method of physically inflicting dynamic forces was somewhat crude, Adams did see the need of studying the limitations of the platform before use with dynamic subjects.

Human engineers in the past have utilized static calibration in estab-

lishing a relationship between the values obtained from the platform and the actual force exerted by the subject. Intuitively, this procedure appears to be erroneous because the human is not a static object; he is internally dynamic even when he is motionless. When a subject on the platform starts to move his body to perform a task, he has to exert force to overcome the inertia and possible frictional effects of the platform plus the weight of his body if force sensing is to occur. If inertia is found to be instrumental in determining a calibration factor, then static calibration becomes somewhat unrealistic. Why use static calibration on a device which evaluates a dynamic being?

The following hypotheses were formulated:

- I. For a given force, dynamic forces will have a lower recorded value than static forces.
- II. The platform transformers will respond linearly to frequencies of force application from 5 CPM to 300 CPM.
- III. Simulated dead weight loads of 100 and 200 pounds will have a negligible effect upon the calibration and operation of the platform.
- IV. The smallest detectable dynamic force in any spatial axis of the force sensing system will be .05 pound.

## METHOD

## Design

Physiological, mechanical, and electrical concepts were united in purpose for developing design criteria. Each part or system was first studied separately in order to develop an understanding of the basic system as they relate to this experiment. This knowledge was then applied to an actual instrument design.

Physiological. The first consideration was: Just how many repetitions or cycles per minute can a human produce without becoming inconsistent? For example, if a typist can type 120 words per minute with five letters per word, she would be performing at a rate of 600 work-cycles per minute. Each cycle would correspond with a finger movement to a key, striking the key and returning to a reset position. An upper limit then would be an instrument which will perform at a rate of at least 600 cycles per minute. A lower limit would necessarily approach zero since a human can perform at a rate which approximates zero.

The next consideration was to find an instrument which operated at a constant speed and could be varied by an external means within this theoretical range (0 to 600 CPM).

The instrument also would have to overcome the inertial effects of a simulated subject weight (dead weight) of 200 pounds and exert a known force upon the sensing units (LVDT) of the force platform.

Electrical. As was stated previously, the need was to design an instrument which would apply an intermittent non-variable force on the platform and accomplish this over a range of frequencies; an electric motor appears to best fulfill these specifications. With the addition of a rheostat to the electrical system (DC motor), the motor speed could be varied at will. This

motor-rheostat combination would greatly sophisticate force-frequency input as compared to the motor-speed reducer-yoke-input shaft-spring system as used by Jacobson. The feasibility of this approach depended on finding a motor which would operate at a constant speed, use DC voltage, and maintain a high torque. The DC voltage will be thus constant.

Once a certain speed was predetermined by adjustment of the rheostat, it should not change. With this fact in mind an AC-DC, universal series-sound motor was selected due to its ability to maintain a constant speed under load if there is a constant voltage power supply.

Another design which was considered in selecting a motor was that of using a synchronous motor coupled to a variable frequency power supply. Two major disadvantages were encountered in this approach. First, the maximum speed attainable with a synchronous motor was approximately 100 CPM. This factor does not meet the hypothetical limits of 0 to 600 CPM as determined in the physiological considerations. Second, although the less than five per cent error in speed control for synchronous motor would be less than the five to ten per cent error with a universal motor, the difficulty involved in building the variable frequency power supply tended to offset this positive feature.

A stroboscope was selected over a tachometer for CPM determination because a stroboscope was readily available for use through the Industrial Engineering department and a tachometer would have been quite hard to fix to the motor shaft for CPM determination.

A stroboscope, through intermittent flashes of light, makes a revolving body appear stationary. When this stationary effect has been obtained, a

direct CPM reading can be taken from a dial indicator on the stroboscope. Thus by setting the stroboscope to a preselected CPM and adjusting the rheostat, the speed of the motor can be adapted to the flashes of the stroboscope and consequently the desired CPM.

Mechanical. With the capability of maintaining a constant motor speed already determined, the most practical approach to force creation was through revolving a known weight within the plane of the force platform's transformers. This was accomplished by affixing a small diameter rod (9/32 inch) to the shaft of the motor such that its weight was evenly distributed over the surface of the platform. The even distribution was achieved by fastening the rod at its center.

By adding a known weight to one end of the rod and revolving this weight at a known speed, a specific centripetal force was established (Appendix I). Preliminary to performing this procedure, the instrument was run at a preselected speed without the attached weight. This facilitated elimination of any extraneous errors due to uneven weight distribution or any adverse inertial effects due to the weight of the rod. In summary, if the transformers of the platform are set in a null position while the rod rotated but before attaching the weight, then, after the weight was added, the transformers would reflect only how they are affected by this additional force.

The mechanical system has two equations which have applicability to this research. (Appendix I) These are:

$$F = \frac{mv^2}{r} \quad (1)$$

and,

$$T = m(v^2/r + g) \quad (2)$$

where  $F$  = the force developed by centripetal motion when a weight is revolved in a plane parallel to the earth's surface. (pounds)

$m = W/g$  = mass of the body which is revolved, and  $W$  = weight of the revolved body. (pounds)

$v$  = velocity of the revolving body. (radians/second)

$r$  = radial distance from the center of the driving shaft to the center of the weight of the body. (feet)

$T$  = force developed by centripetal motion when weight is revolved in a plane perpendicular to the earth's surface. (pounds)

$g$  = force of gravity. (32.17 feet/second<sup>2</sup>)

These equations give an instantaneous value of force for each spatial axis. The particular equation used depends upon the plane in which the weight is revolved; if revolved in the horizontal plane (parallel to the earth's surface), the frontal and lateral transformers sense a force. When the weight is revolved in the vertical plane and parallel to either the lateral or frontal plane, the vertical and either the lateral or frontal transformer respond, depending upon which of the latter two axes is in the plane of rotation. If revolved in the vertical plane and in a plane bisecting the lateral or frontal planes, all three transformers will respond.

Equations one and two were used to determine the specific weights which would need to be revolved. Since this study was to determine the smallest and largest forces detectible by the platform, these equations gave a means of theoretically producing some of the forces to be encountered. The radius of rotation was preset, as will be explained later, and the frequency range (velocity) has been predetermined in the physiological system study; so to find a range of weights which would produce minimum force of .05 pound and maximum force of 50 pounds various weight values were substituted into equations (1) and (2) until these limits were satisfied.

Practical considerations. The factors considered critical in the physical dimensioning of the calibration instrument were uniformity in the dis-

tribution of weight over each plane of the platform, and the flexibility involved in creating responses in only two of three planes (XY, YZ, or XZ) by simply altering the position of the instrument on the surface of the upper plate.

Two reasons why the size of the instrument should be kept within the limits of the upper plate surface area are stated below. Increasing the radius of swing of the weight will lessen the CPM needed to produce the same force or to maintain a constant force. From equation one it can be seen that an increase in  $r$  will necessitate a decrease in CPM. Therefore by keeping the radius shorter, there is less need for slow CPM. On the other hand, the lower the CPM, the more closely the average working speed of a human is approached. Since this research was to develop a range of operating values and not an average, the last pragmatic approach was rejected. A second consideration was an instrument which could be maneuvered with little physical exertion; this eliminated an instrument which stood much over two or three feet high.

Due to the need for maneuverability, instrument weight should be low. Since the motor selected weighed 7 pounds, the frame material needed to be a sturdy, light material to keep the device under 50 pounds. Aluminum fulfilled these specifications at low cost.

The frame and inner motor support beam were made of five-eighth's inch aluminum plate. (Appendix II) Five-eighth's inch was selected because of its availability, not for its heavy duty characteristics. Three-eighth's inch aluminum, if available, would be sufficient in strength.

The overall dimensions as shown in the drawings of Appendix II were not set and then strictly adhered to in machining the rough material. Since the outer dimensions were required only to be of equal length, machining was done

with forethought only to the final square shape. A second criterion in machining was to reduce the size of the rough sheets such that they would fit within the confines of the upper plate of the platform.

Again, as noted in the drawings in Appendix II, there are two sides of the instrument which contain small holes. Of the three holes drilled in each side, the center hole is a positioning hole; whereas the two outside holes are securing holes. All of the  $3/8"$  diameter holes on each side are unthreaded.

The primary positioning holes (center holes) on the end and side of the calibration instrument's frame (Appendix II, drawing 506) mate with center hole drilled in the upper plate of the platform (Plate I). The hole corresponds with the exact center of the platform so all of the revolutions of the calibration motor will be in a plane with reference to the true center of the platform and necessarily the overall weight of the instrument.

The two securing holes in both the side and end of the instrument's frame have mating holes in the platform also. These mating holes lie on the perimeter of a circle of holes (Plate I). Each hole was drilled at forty-five degree angle increments along this perimeter. In order to stabilize the instrument operation as much as possible, the holes lying on the perimeter of this circle on the upper platform plate were to make the calibration of any two or all three spatial axes more efficient. For example, if the long side of the instrument was placed in the plane of the frontal transformer ( $0^\circ$ ), the readings would be obtained from the lateral transformer and the vertical transformer. Readings would be produced by the frontal and vertical transformers if the instrument were placed with the long side in the plane of the lateral transformer ( $90^\circ$ ). Consequently, if all three planes were to be studied, the instrument would most readily accomplish this if it were placed

in this vertical position of the  $45^{\circ}$  or  $135^{\circ}$  lines. It follows that the frontal and lateral transformers could be studied together if the short side (22.859 inches) of the instrument were to be secured in any of the angle positions mentioned above. This is accomplished because the plane made by the revolving rod will be parallel to the upper plate of the platform; this makes all placings of the base of the device equal in the final effect desired.

In studying forces in all three planes an instrument of this design needed only to be placed in one or two of a possible eight positions. If a range of force applications for each axis was to be found, the basic procedure amounted to rotating the weight in a vertical position (plane perpendicular to the platform surface) in both the  $90^{\circ}$  and  $0^{\circ}$  planes. The vertical forces were recorded each time in relation to either the lateral or frontal forces, depending on which of the latter two transformers was in the plane of swing.

After determination of specific weights to revolve (.10, .25, .50, and 1 pound) to obtain the desired forces, and the overall dimensions limiting the radius of swing, the final design criterion for a motor selection was torque. To find a motor capable of overcoming the torque involved, the torque to be created had to be calculated. The maximum torque to be encountered was 9 inch pounds (radius X weight = 9 inches X 1 pounds). From this number a value of horsepower (hp) could be found by multiplying by the cycles per second. This figure indicated that a  $1/20$  hp motor would be sufficient for this application. In order to overcome internal motor losses, frictional losses, inertial losses and the pendulum effect in the vertical plane a motor must be selected which has sufficient hp to maintain the constant speed needed; intuitively a safety factor of four was used to adjust the hp needed to  $1/5$  hp.

When the weight is spinning in the vertical plane, the effect of gravity will tend to increase or decrease the speed of revolution depending upon the half-cycle of operation. Although formula 2 has compensated for the effect of gravity, the errors encountered in maintaining a constant motor speed will be somewhat greater than when the weight is spun in a horizontal plane. The only way of reducing these errors appreciably is to use an extremely large hp motor. But, for this experiment, the 1/5 hp motor used proved quite sufficient. The only problem foreseeable was establishment of constant motor speed with the stroboscope.

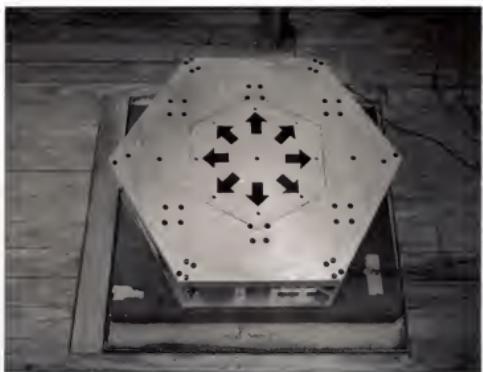
In summary, an instrument was designed which could inflict a measurable force at intermittent intervals upon the platform's transformers.

This instrument consists of a rheostat controlled, fractional horsepower AC-DC motor, mounted on a middle brace within a square frame such that a rod can be revolved directly over the center of the platform. This instrument meets the requirements of even distribution of weight over the platform's surface.

DESCRIPTION OF PLATE I

Illustration showing the center hole and corresponding circle of holes  
in the upper plate of the platform.

PLATE I



### Experimental Procedure

Step one of the experimental design involved calibrating a two-channel oscillographic recorder (Texas Instruments, model P2CBH). Actually, calibration was performed upon a carrier amplifier (model number 10 - 198661 - 1) which is the integral part of the recorder for converting the responses from the LVDT's into impulses which can be recorded graphically. Plate II illustrates the control panel for the carrier amplifier used in calibrating its internal electronic components.

The first sequence in preparing the amplifier-recorder for force detection was to turn the attenuation dial to OFF. Next, the position dial was used to adjust the pen position on the graph to a center position.

After centering the pen on the paper, the attenuation dial was adjusted to CAL(+2cm). The deflection of the pen from the initial location should be 2 cm while in this attenuation position. If not, the gain dial was used to manually position the pen at 2 cm from the center.

Following this procedure, the attenuation was returned to the OFF position--the pen should again be at the center of the recording paper. If it wasn't--the position dial was adjusted. If it was--the attenuator was turned to 10K, and the screwdriver-adjustable, C-balance screw was used to re-center the pen position.

The sequence followed from this point was that of lowering the attenuation dial scale factor and adjusting either the C or R balance to zero-position the pen, until the pen did NOT balance. Then by backing up one attenuation scale factor and adjusting the R or C balance, the oscillograph was ready for recording.

Step two of the experimental procedure was begun by placing the calibration instrument upon the platform in the horizontal position (spins weight

parallel to the platform's surface). The device was then positioned parallel to the frontal transformer (or lateral) and aligned by eye to the center hole in the upper plate of the platform. When the alignment appeared to be true, a tapered positioning pin (part number 501) was fitted into the frame of the instrument, threaded through to the platform plate, until it achieved a snug fit in the plate's center hole. Next, the instrument was slightly moved clockwise or counterclockwise, depending upon the orientation of the outer two securing holes relative to the holes in the upper plate. Here 20-3/8 bolts are inserted in the holes in the frame and secured in the threaded holes in the platform. These two bolts must be tightened securely to prevent any adverse vibrations or inertial effects that might arise during operation.

Step three of the experimental procedure involved turning the motor on. When the CPM wanted was achieved, the recording instruments were re-balanced to a zero position by using the R and C balance adjustments. Before balancing the instrument electronically, the weight distribution of the instrument was evened over the surface of the platform by using the weight as shown in Plate III. Simultaneously a simulated zero-weight position of the pen on the recorders was attained through the balance dials.

Uniform addition of either 50 or 150 pounds of deadweight to the platform surface, being mindful of a zero strip-chart recording at all times, was the fourth step in the procedure. This step renders all apparent extraneous errors to a non-significant magnitude. In other words everything was set to zero in preparation for the actual calibration with these simulated weights.

Step five of the procedure entailed the actual addition of the weight to the rod, after shutting off the motor. Then, after the weight was attached, the motor CPM was again set to the calibrated value. At this point the re-

DESCRIPTION OF PLATE II

Photograph shows the recording equipment used in this experiment.

Illustration of control panel for the carrier amplifiers is shown below the photograph.

PLATE II

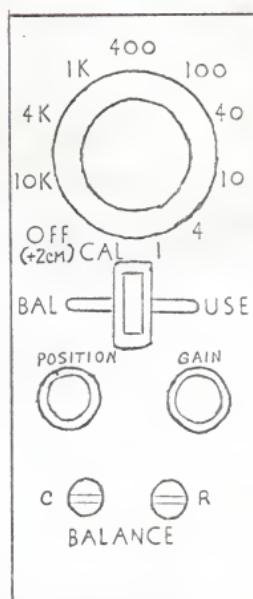
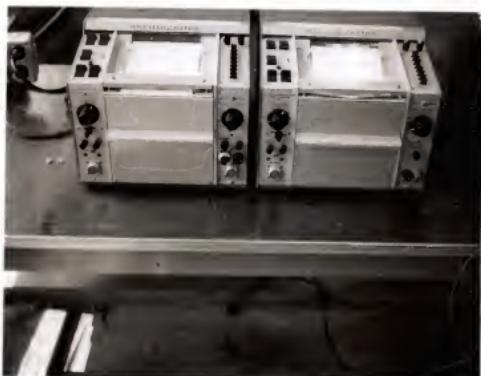


PLATE III



DESCRIPTION OF PLATE III

Photograph showing the counterbalancing weight being placed such that it will equally balance the weight of the motor and center beam. This placement will produce a couple upon the center pivot of the platform.

corder should be recording an almost perfect sinusoidal wave; that is, if the procedure given above was followed correctly. Interpretation and evaluation of these sine waves will be described in the Evaluation section of this METHOD chapter.

An additional procedure to step five occurred when the equivalent of a .01 pound weight was to be revolved. In this case both the .10 and .25 pound weights were attached to the rod. Their respective distances from the center of the rod were varied depending upon the incremental weight desired. To counterbalance the two weights, the .10 pound weight was placed at 8.936 inches from center and the .25 pound weight was placed at 5.656 inches from center. Then, to get the effect of adding a .01 pound weight, the .1 pound weight was moved to 8.906 inches from center.

Step six was used to establish the static calibration constants for this research. The historical method of fixing a string to the platform in direct line with one of the transformers, laying the string over a pulley, and attaching the known weight to the other end of the string was employed. The above mentioned method was used for the lateral and frontal transformers. A response was obtained from the vertical transformer by simply placing the known weight upon the upper plate of the platform.

## METHOD

## Construction

The drawings of Appendix II are the only manufactured parts needed in constructing and operating the calibration instrument. Drawings 500 through 504 can be assembled in any order necessary to achieve a final structure as shown in drawing 506.

After completion of the basic frame assembly, the shaft coupler (drawing 505) was fastened to the motor shaft by means of a  $\frac{1}{4}$ -20 set screw. The purpose of the coupler was to provide a means of coupling the shaft of the motor to the rod to be revolved. The shaft coupler and motor are then mounted on the middle brace of the frame. Plate IV shows how the instrument will look after complete assembly.

The 9/32 drill rod will not deflect appreciably when loaded with the maximum torque applied (9 inch-pounds). The rod was made 18 inches long in order to leave a clearance between the weight fixed to it and the frame of the calibration instrument. The rod was then fastened to the shaft coupler at the rod's mid-point by means of a  $\frac{1}{4}$ -20 set screw in the end of the shaft coupler.

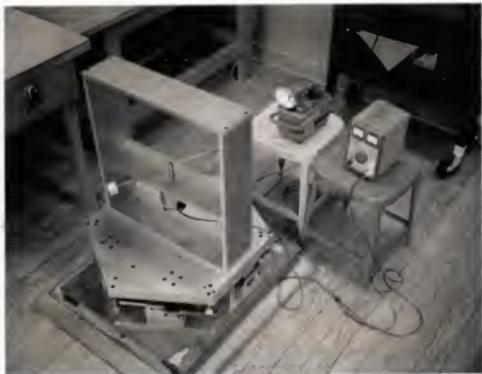
The other manufactured parts used in the actual calibration were the weights as shown in drawings 507 through 511. The two pound weight (which was not used in the dynamic calibration due to the large forces created) and the one pound weight were machined from cylindrical steel bars. Before a final weight for each had been attained within a plus 40 grams, two set screws ( $\frac{1}{4}$ -20) were positioned such as to be at right angles to each other in the cylinder wall. Next, a center hole was drilled in these cylindrical weights to facilitate their sliding over the rod mentioned above. The drilling of the set screw holes and center hole would leave the weight at or above the desired weight; thus, if necessary, to render the weight as close to the de-

DESCRIPTION OF PLATE IV

Photograph showing the complete assembly of the calibration instrument including the shaft coupler, rod and the one-half pound weight.

The stroboscope and the battery eliminator (DC power supply with rheostat control of voltage) are shown in the background.

PLATE IV



sired weight as wanted, the cylinder could then be turned down more in a lathe. The cylinder-weights were finished machined to plus or minus 5 grams. The .50, .25, and .10 pound weights were made from aluminum and machined in accordance with the above described procedure.

## METHOD

## Evaluation

The necessary prerequisite to evaluating graphical data for this report was an analysis of its content and character. Specifically, an interpretation of the recordings or responses was followed by an analysis of the performance of the platform.

The data to be interpreted for this research appears in the form of a sinusoidal wave on the strip-chart recording paper (Plate V, part (a)). One cycle of a sine wave is equivalent to one revolution of the weight of the calibration instrument. The reason for having a sine wave was due to the position of the weight in revolution relative to the transformer being actuated. When the rod with attached weight was parallel to the core of the responding transformer and directly above it, the recording pen was at the maximum amplitude point in the positive half-cycle of a sine wave. Conversely, when the weight was directly opposite and parallel to the same transformer, the pen was at a maximum amplitude in the negative half-cycle of a sine wave.

A sine wave recording was obtained with each combination of experimental conditions. The variables in mention are the revolved weight, simulated deadweight, CPM, and axis (lateral, frontal, vertical) used.

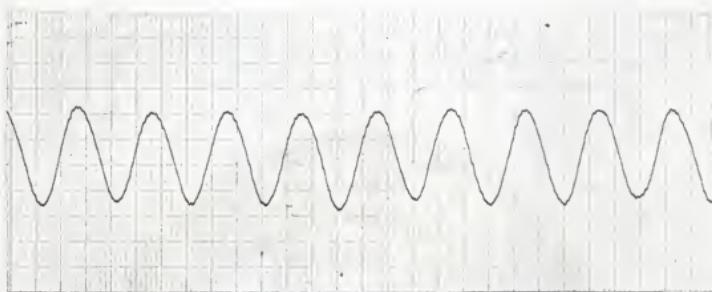
Initially, the responses recorded on the oscillographic recorder were studied for a sinusoidal waveform (Plate V), and then the maximum amplitude of these sine waves was measured. If the recordings were in a non-sinusoidal pattern, such as shown on the second chart of Plate V, the transformers were investigated as to possible malfunctions such as sticking, bad wire connections, and misalignment of ball-end cores. Conversely, finding a sine wave was indicative of the expected response and was followed by graphical recording of the strip chart value (mm-deflection).

DESCRIPTION OF PLATE V

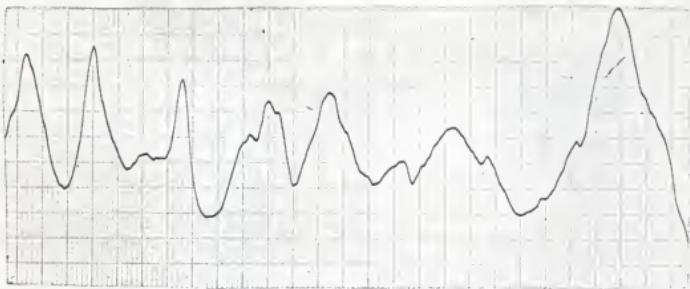
- (a) Top illustration shows the sinusoidal response obtained when the platform was responding correctly.
- (b) Bottom illustration shows improper functioning of the platform.

## PLATE V

(a)



(b)



Interpretation of the strip chart recording for static calibration was similar to interpretation of the dynamic responses, only that once the recording pen had deflected it would not vary. This deflected value was then recorded as the static calibration constant for the particular force applied.

Accumulation of these two types (dynamic and static) of graphical data is illustrated in Appendix IV. Actual interpretation of these graphs follows in the RESULTS section of this report.

## RESULTS

The four criteria established as indicators of the response characteristics of Hearn's platform were: (1) effect of added mass to the platform, (2) force-frequency response range, (3) dynamic calibration as opposed to static calibration, and (4) minimum force detectable.

For static calibration, the effect of 0, 50, 100 and 150 pounds of dead-weight on the three axes was investigated; see Figures 1 to 4 for the lateral axis, 6 to 9 for the frontal axis and 11 to 14 for the vertical axis. The four graphs are plotted on Figures 5, 10, and 15. After a run with each dead-weight the carrier amplifier was rebalanced for the new run. The force values within a run were quite consistent; the differences in slopes of the lines are attributed to differences in balancing sensitivity rather than an effect of deadweight.

When comparing static to dynamic calibration, it was found that specific statements relating the two methods must necessarily be restricted to each axis separately. For the lateral axis it was found that static was no different than dynamic when inflicting forces (Graphs 5 versus 17). The frontal axis did show a slight discrepancy in the two methods with output being recorded slightly higher for a given dynamic input than the static forces (Graphs 10 versus 19). The largest discrepancy occurred in the vertical axis where the dynamic forces indicated twice the output from the transformer for a given input than the static force (Graphs 15 and 21). Graph 23 combines Graphs 17, 19, and 21; Graph 24 combines 18, 20, and 22. The data for all three axes were averaged and compared (See Graph 25). A general statement to be drawn from this graph would be that dynamic forces have a slightly higher degree of sensitivity associated with them.

The platform will respond to an experimentally determined minimum CPM

of 30. For reasons of safety the maximum output became 540 CPM. In neither of these cases was the actual limits of the platform approached. By manually inflicting a timed force, it was found that the actual lower limit does approach zero. The upper limit was restricted for fear of damage to the instrument.

When considering the smallest detectable force each transformer must be considered separately. The minimum detectable force in the lateral plane was on the order of .02 pounds (Graph 18). The frontal axis indicates a somewhat similar minimum response of .025 pounds (Graph 20). The vertical transformer will respond to forces of .01 pounds; thus it is the most sensitive.

#### DISCUSSION

Each of four hypotheses formulated for this research was either invalidated or numerically underestimated. In three of the cases there was a marked difference between predicted and actual results. This fact was not necessarily an incorrect experimental design, but a lack of foresight on the part of the experimenter of the degree of sophistication of design of the Hearn platform.

#### Static and Dynamic Calibration

The difference between static and dynamic calibration as found by other researchers in the past was small for two of the three axes for this experiment (no difference for the lateral axis and static approximately 10 per cent less sensitive for the frontal axis). For the vertical axis, static calibration gave only 50 per cent of the output as dynamic calibration. This conflicts with Jacobson's results where static calibration was more responsive than dynamic.

Past platforms have been adversely effected by resonance (Greene's platform as modified by Jacobson) when operating at frequencies of 150 CPM or more. This fact along with the apparent effect of inertia have been major reasons for not attaining similar responses for equivalent dynamic and static force applications. Graph 15 versus Graph 21 and 22 indicates that the method of vertically inflicting force might have been in error due to the inability of this curve to pass through zero. A .5 pound static force inflicted gave an output of approximately .8 mm; when a .25 pound weight was used an output of 4.5 mm was recorded for a .5 pound input. When two almost balanced weights were used to inflict a .01 pound weight an output of 1.7 was recorded for a .5 pound input. Thus the unexpected value for dynamic vertical calibration may be caused by an inherent flaw in the calibration technique.

rather than the platform design.

A generalization to be drawn from the experiment would be that dynamic calibration was only slightly more sensitive than static calibration and in normal operation static calibration would be sufficient.

#### Frequency Range

Although the range of frequency input for this research was restricted to an experimental range, these limitations were exceeded in order to obtain the frequency response limits of the platform. The lower limit of the frequency was restricted by the low speed of the motor. This minimum was on the order of 30 CPM. Conversely, the upper limit of 540 CPM was restricted by the experimenter for fear of damage to the instrument (platform). This upper limit, although not a maximum of the platform, must be considered as a practical maximum of the calibration instrument due to the forces involved at these higher CPM's.

#### Effect of Added Mass

The design of this experiment was modified for this phase of the research. Due to the sensitivity involved when working with Hearn's platform, the deadweight conditions were altered from the initial 0, 100, and 200 pound conditions to 0, 50, 100, and 150 pounds.

Since the plots of all the data obtained were linear, the addition of deadweight was considered to have no effect upon the sensitivity of the platform. If there had not been a linear relationship for each set of data, then it would have been concluded that the platform was functioning improperly, thus rendering much of the data invalid. Intuition indicates that added mass would cause added inertia which in turn would cause a decrease in the

responses obtained. The linear relationship indicates proper functioning for the platform; thus, the deadweight conditions do not effect the functioning of the platform.

#### Minimum Force Sensitivity

Researchers in the past have cited the sensitivity in measuring a heartbeat as a criterion for minimal platform response. This standard may well have been a good measure in the past, but with the present platform the heartbeat becomes a quite large force to contend with in data analysis. The actual force value for a human heartbeat was approximately .5 pound in the frontal axis. If this value is then compared to the minimum response obtained upon the present platform of .01 pound, the past standard becomes a large force. The above mentioned minimum force of .01 pound was obtained in the vertical plane only. The minimum for the other two axes was .02 and .025 pounds.

The reason for not attempting to induce a smaller force in any of the axes was due to the human error involved in counterbalancing the .1 and .25 pound weights on the rod such that the difference in their masses when revolved would produce an incremental force upon the platform. The adjustment of the difference in the moment arms of the weights on the rod to obtain the .01 pound force was measured with a vernier calipers. With this amount of accuracy needed, there was little room left for a finer adjustment.

#### Other Considerations and Effects

Another aspect was the experimental determination of the resonant frequency of the platform. The resonance value was established by tapping the

side of the upper plate lightly with a small hammer and counting the number of oscillations obtained per unit time on the strip chart recorder. Resonance for the frontal (X-axis) exists at 3500CPM, whereas resonance for the lateral axis (Y-axis) was somewhat less at 3250 CPM. A distinct resonant frequency could not be attained from the vertical transformer.

With many of the aforementioned characteristics of the platform as criteria for future research, recommendations for designing a dynamic calibration instrument with a sensitivity which more closely approaches the sensitivity of the present platform follows: (1) a low voltage motor should be selected which varies from less than one CPM to more than 500 CPM; the slower speeds are the most important; (2) a variable-frequency constant voltage power supply should be designed to give close control over the voltage input to the motor; it should have graduated settings for specific CPM's desired; and, (3) a means of precisely determining (.001 inch) the distance from the center of the motor shaft to the center of inertia of the revolved weight should be used.

Since a future is forecast for using the force platform in studying minute physiological differences in individuals, a means of precision calibration must be established.

#### SUMMARY

Hearn's platform was found sufficiently sensitive to measure a .01 pound force in the vertical plane while the lateral and frontal axes were able to detect forces of .02 and .025 pounds, respectively. When a deadweight of 50, 100, or 150 pounds was added to the platform the sensitivity was not appreciably affected.

Operational frequencies for the platform span a range beyond the physical capacities of a human. Calibration problems encountered at low speeds caused 30 CPM to be the lower limit of frequency input to the platform; the platform was satisfactory down to that speed. The practical maximum of the calibration instrument for force infliction was attained at 540 CPM. The resonance or natural frequency was found to be 3500 and 3250 CPM for the frontal and lateral axes, respectively. Resonance could not be experimentally determined for the vertical axis, but it was felt that this value would be quite high.

No difference was found between static and dynamic calibration for the frontal and lateral axes. Apparent errors in the vertical method of dynamically inflicting force caused this axis to show a difference between static and dynamic of 50 per cent; the error is considered to reside in the calibration technique not the platform. The negligible difference between static and dynamic force infliction for the lateral and frontal axes attests to the precision of the sensing units and design of the platform.

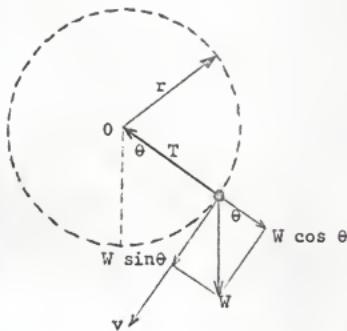
Need was seen for the development of more sophisticated calibration equipment to compensate for the precision designed into the present platform. Future designers of dynamic calibration instruments are urged to construct a variable frequency power supply suitable for low speed applications. A low speed, low voltage motor must be selected for wider, more constant frequency ranges. This development, it is hoped, will improve future use of the platform in studying the physical exertions of the human.

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**APPENDIX I**

The illustration below is a diagram of the calibration instrument's rod of radius  $r$  with affixed weight  $W$  being revolved in a vertical circle. It must be remembered that while revolving a weight in this vertical position its speed will increase on the downward swing and decrease on the upward swing.



The two forces acting upon this weight are the tension in the rod  $T$ , and the weight of the body  $W = mg$ . If the weight is resolved into its normal and tangential components as shown in the above illustration, a summation of forces in the X and Y directions will yield resultant normal and tangential forces of  $T - W \cos \theta$ , and  $W \sin \theta$  respectively. Then by applying Newton's second law,  $F = ma$ , to find the normal acceleration  $a(N)$  as follows:

$$a(N) = \frac{F(N)}{m} = \frac{T - W \cos \theta}{m} = \frac{v^2}{r} \quad (\text{radial acceleration})$$

and, solving this equation for  $T$ , where  $W = mg$ ,

$$T = m(v^2/r + g \cos \theta)$$

a force for calibration can be determined; if it is considered that the maximum force attainable on the vertical transformer will occur when the weight is at the lowest point of its path. At this point of concern,  $\theta = 0$ ,  $\sin \theta = 0$ ,  $\cos \theta = 1$ . Also, the tangential force will equal zero; reducing the tangential acceleration to zero likewise. Thus the magnitude of the tension

in the rod and consequently the force applied to the vertical transformer at this point will be:

$$T = m(v^2/r + g)$$

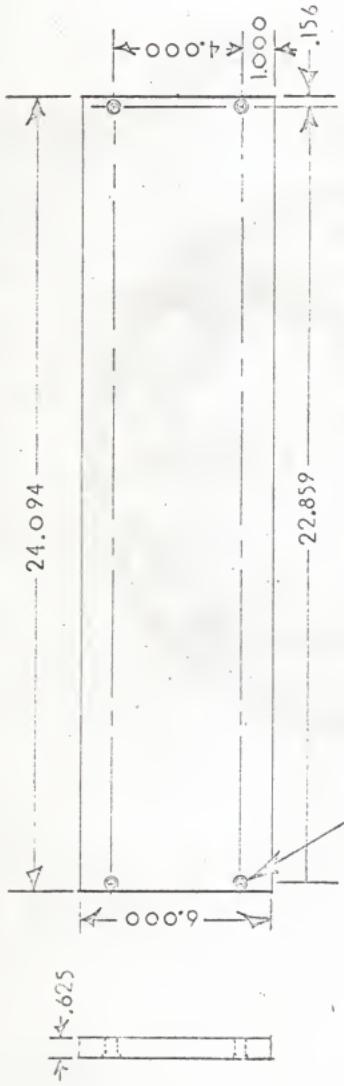
For solution of the force in the horizontal plane the effect of gravity can be simply ignored because gravity is everywhere equal upon the weight during each cycle of operation. With this fact in mind, the above equation for vertical motion resolves to:

$$T - mg = F = mv^2/r$$

which is the instantaneous force (normal force value at any time on a body being revolved in a horizontal plane).

APPENDIX II

24.094



$\frac{3}{8}$  DRILL THRU 4 HOLES  
C' BORE FOR  $\frac{3}{8}$  - 16 SOCKET  
HEAD SCREW

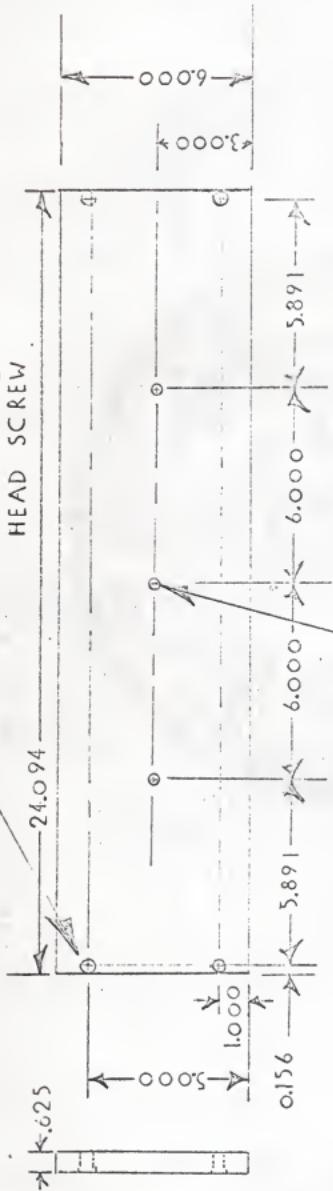
TOLERANCE UNLESS SPECIFIED  $\pm 0.005$

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TERMINAL TESTS FOR THE DEPARTMENT OF INDUSTRIAL ENGINEERING, KANSAS STATE UNIVERSITY, LAWRENCE, KANSAS, ARE CONDUCTED IN ACCORDANCE WITH THE STANDARDS FOR TERMINAL TESTS FOR THE DEPARTMENT OF INDUSTRIAL ENGINEERING, KANSAS STATE UNIVERSITY, LAWRENCE, KANSAS, AS PUBLISHED BY THE AMERICAN SOCIETY FOR TESTS AND MATERIALS, IN THE STANDARD FOR TERMINAL TESTS FOR THE DEPARTMENT OF INDUSTRIAL ENGINEERING, KANSAS STATE UNIVERSITY, LAWRENCE, KANSAS, AS PUBLISHED BY THE AMERICAN SOCIETY FOR TESTS AND MATERIALS, IN THE STANDARD FOR

Project FORCE			
Material	Platform CALIBRATION	Scale	Date
ALUMINUM	1" = 4"	3-1-67	App'd
No. Req'd	1	Drawn By	N.D.GRANGER
Part Name	FRAME MEMBER - LONG	Part No.	500

$\frac{3}{8}$  DRILL THRU 4 HOLES  
 C'BORE FOR  $\frac{3}{8}$ -16 SOCKET  
 HEAD SCREW

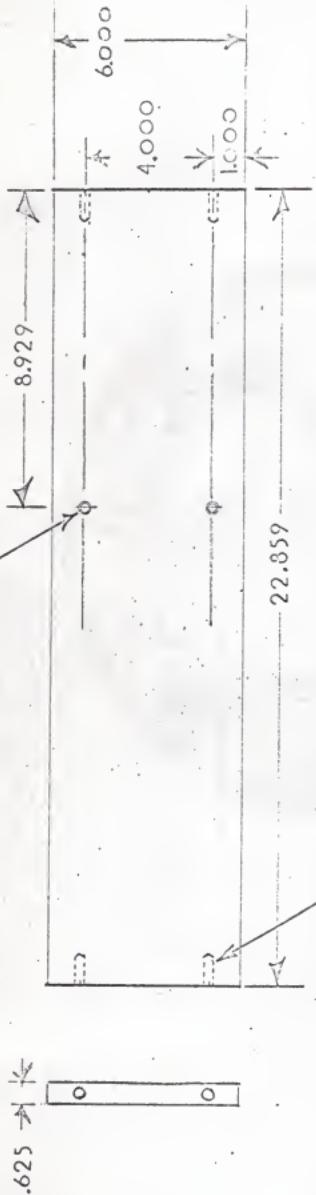


$\frac{3}{8}$  DRILL THRU 3 HOLES

TOLERANCE UNLESS SPECIFIED ±0.005  
 DEPT. OF INDUSTRIAL ENGG.  
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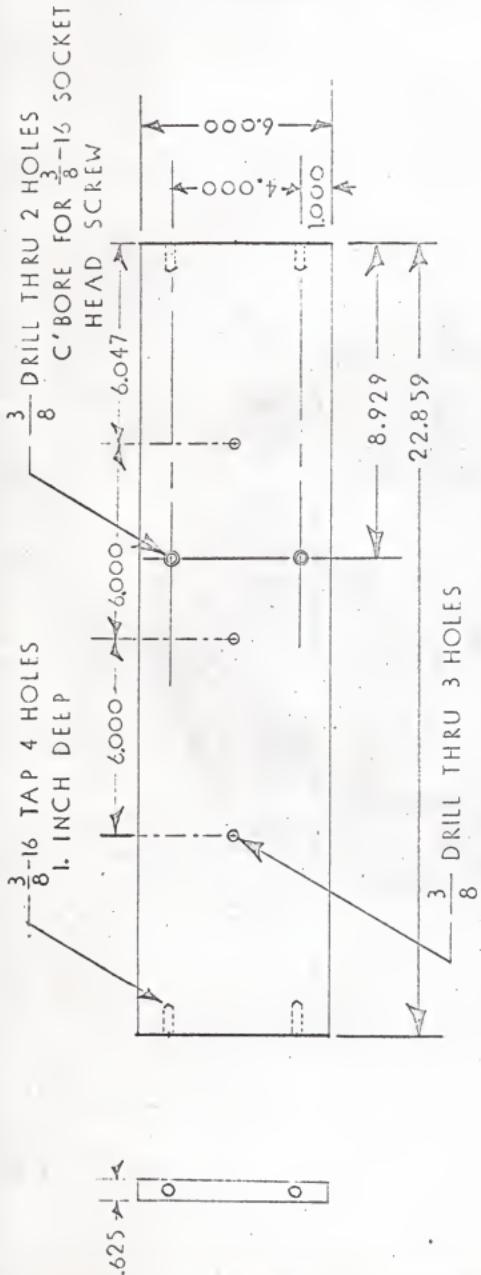
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Material ALUMINUM	Scale 1" = 4"	Date 3 - 25 - 67	
No. Req'd 1	Drawn BY N.D.GANGER	App'd	
Part Name HORIZONTAL SECURING PLATE	Part No. 501		

$\frac{3}{8}$  DRILL THRU 2 HOLES  
C'BORE FOR  $\frac{3}{8}$ -16 SOCKET  
HEAD SCREW



TOLERANCE UNLESS SPECIFIED ±0.0005  
DEPT of INDUSTRIAL ENGG.  
KANSAS STATE UNIVERSITY

Project FORCE		PLATFORM CALIBRATION	
Material	Scale	Date	
ALUMINUM	1" = 4"	3-1-67	
No. Reg'd	1	Drawn By	Appd
Part Name:	N.D.GRANGER		
FRAME	FAC. SIRE R.	SCHERI	Part No.
			SO2



TOLERANCE UNLESS SPECIFIED  $\pm 0.005$

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Project

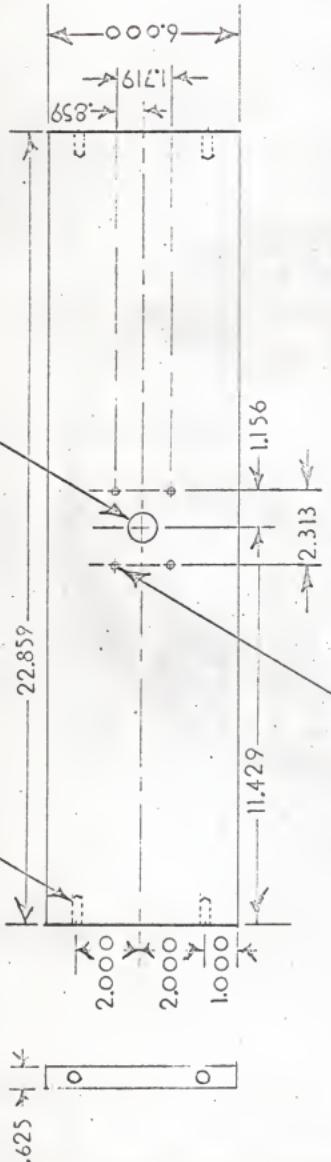
FORCE PLATFORM CALIBRATION

Material	Scale	Date
ALUMINUM	1" = 4"	3-1-67

No. Req'd	Drawn By	App'd
1	N.D.GRANGER	

Part Name	Vertical Securing Plate	Part No.
		503

$\frac{3}{8}$ -16 TAP 4 HOLES  
1. INCH DEEP



#20 DRILL THRU 4 HOLES

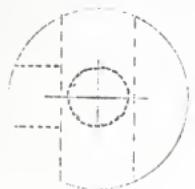
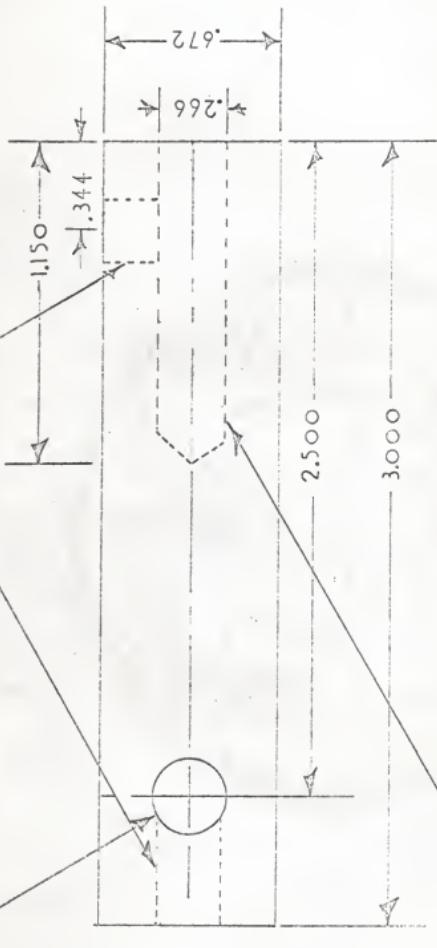
TOLERANCE UNLESS SPECIFIED  $\pm 0.005$   
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Project FORCE PLATFORM CALIBRATION		Date	App'd
Material	Scale 1" = 4"	Drawn By	
ALUMINUM		N.D.GRANGER	
No. Req'd	1		

Part No.  
504

$\frac{1}{4}$  - 20 TAP 2 HOLES FOR  
SOCKET HEAD SET  
SCREW

$\frac{9}{32}$  DRILL THRU 1 HOLE



TOLERANCE UNLESS SPECIFIED ± 0.005

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KANSAS STATE UNIVERSITY

Project FORCE PLATE FORM CALIBRATION		Date	App'd
Material STEEL	Scale $2'' = 1''$	3-1-67	
No. Req'd	Drawn By	H.D.GRANGER	
Part Name SHAFT COUPLER	Part No. 505		

MEMBER-  
SHORT

FRAME  
MEMBER -  
LONG

HORIZONTAL  
SECURING  
PLATE

MOTOR

MOTOR  
MOUNTING  
PLATE

ROD

SHAFT  
COUPLER

.625<sup>+.000</sup>  
-5.429<sup>-.000</sup>

22.859

23.369

12.047

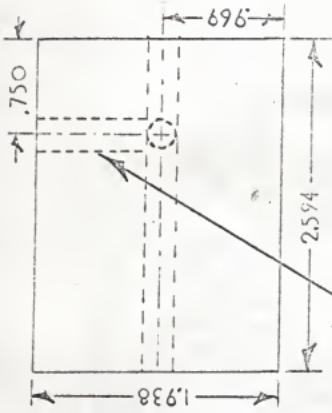
24.094

TOLERANCE UNLESS SPECIFIED  $\pm 0.005$   
DEPT. of INDUSTRIAL ENGG.  
KANSAS STATE UNIVERSITY

Project FORCE PLATFORM

Material ALUMINUM Scale  $1'' = 6''$   
No. Req'd 1 Drawn BY N.D.G.RANGER Date 3-1-67  
App'd

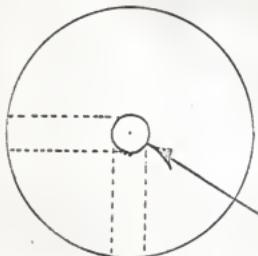
Part Name ASSEMBLY DRAWING Part No.  
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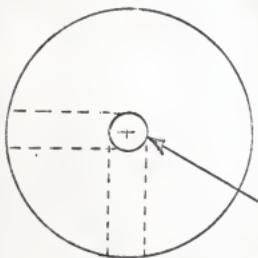
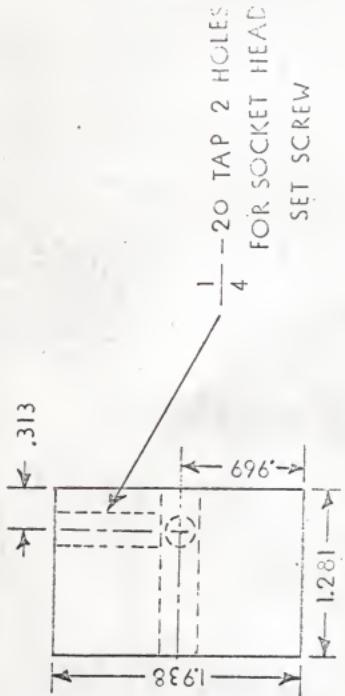
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**KANSAS STATE UNIVERSITY**

Project PLATE FORM CAULDRON			
Material STEEL	Scale FULL	Date 3-27	
No. Req'd 1	Drawn BY N.D.GRANGER	App'd	
Part Name TWO POUND		Part No. 507	
CYLINDRICAL WEIGHT			

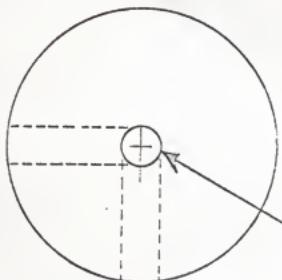
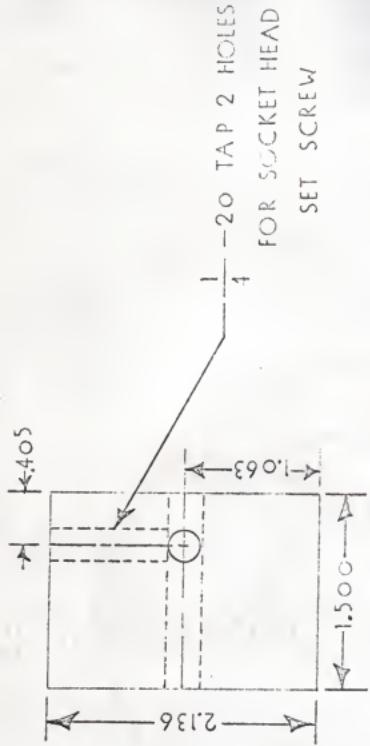


TOLERANCE UNLESS SPECIFIED ± .005  
**DEPT. of INDUSTRIAL ENGG.**  
**KANSAS STATE UNIVERSITY**

Project FORCE PLATEFORM CALIBRATION			
Material	Scale	Date	App'd
STEEL	F. T.L	4-2-7	
No. Req'd	1	Drawn by	
Part Name	ONE POUND	Part No.	503
CYLINDRICAL WEIGHT			



$\frac{9}{32}$  DRILL THRU 1 HOLE



TOLERANCE UNLESS SPECIFIED  $\pm .005$

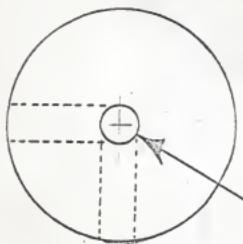
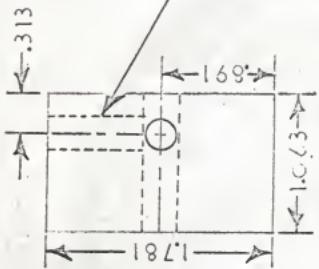
*DEPT. of INDUSTRIAL ENGG.*

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Project FORCE PLATE FORM CALIBRATION

Material	Scale	Date
ALUMINUM	FULL	1-2-77
No. Req'd	1	Drawn BY N.D.GRANGER

Part Name	ONE - HALF PCUN. CYLINDRICAL	Part No.
	Weight	509



$\frac{9}{32}$  DRILL THRU 1 HOLE

$\frac{1}{4}$  -20 TAP 2 HOLES  
FOR SOCKET HEAD  
SET SCREW

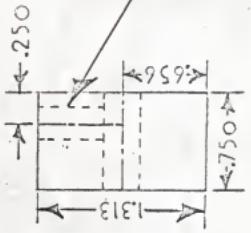
TOLERANCE UNLESS SPECIFIED  $\pm 0.005$

DEPT. of INDUSTRIAL ENGG.  
KANSAS STATE UNIVERSITY

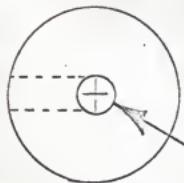
Project  
FORCE PLATFORM CALIBRATION

Material	Scale	Full	Date
ALUMINUM			1-2-77
No. Req'd	1	Drawn By N.D.GRANGER	Appd

Part Name QUARTER FOUND Part No  
CYLINDRICAL WEIGHT 510



$\frac{1}{4}$  - 2C TAP 1 HOLE  
FOR SOCKET HEAD  
SET SCREW



$\frac{9}{32}$  DRILL THRU 1 HOLE

TOLERANCE UNLESS SPECIFIED ± 0.005  
DEPT. of INDUSTRIAL ENGG.

KANSAS STATE UNIVERSITY

Project FORCE PLATE & CALIBRATION

Material	Scale	Date	Part No.
ALUMINUM	FULL	2-77	
No. Req'd 1	Drawn By N.D.GRANGER	App'd	\$11

APPENDIX III

TABLE 1

CALCULATED VALUES OF FORCE<sup>1</sup>  
FOR HORIZONTAL ROTATION<sup>2</sup>

CPM	(Pounds) WEIGHT				
	$\frac{1}{100}$	$\frac{1}{10}$	$\frac{1}{4}$	$\frac{1}{2}$	1
50	.0064	.0639	.1597	.3194	.6387
100	.0256	.2555	.6387	1.2780	2.5550
150	.0575	.5749	1.4372	2.8744	5.7490
200	.1022	1.0220	2.5550	5.1100	10.2200
250	.1597	1.5970	3.9922	7.9844	15.9690

1 These values were calculated by using the centripetal force equation.  

$$(F = \frac{4\pi^2 f^2 W_r}{g})$$

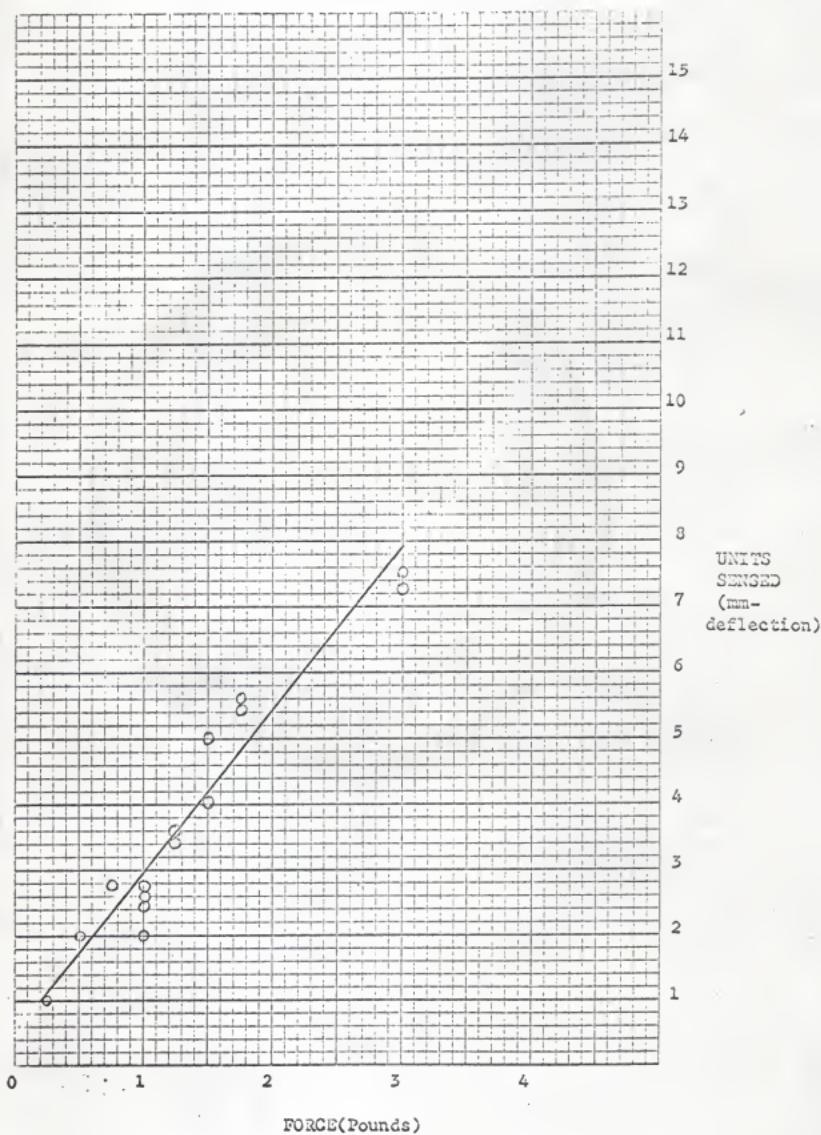
2 Vertical forces are equal to the calculated horizontal force plus the weight revolved.

APPENDIX IV

GRAPH 1

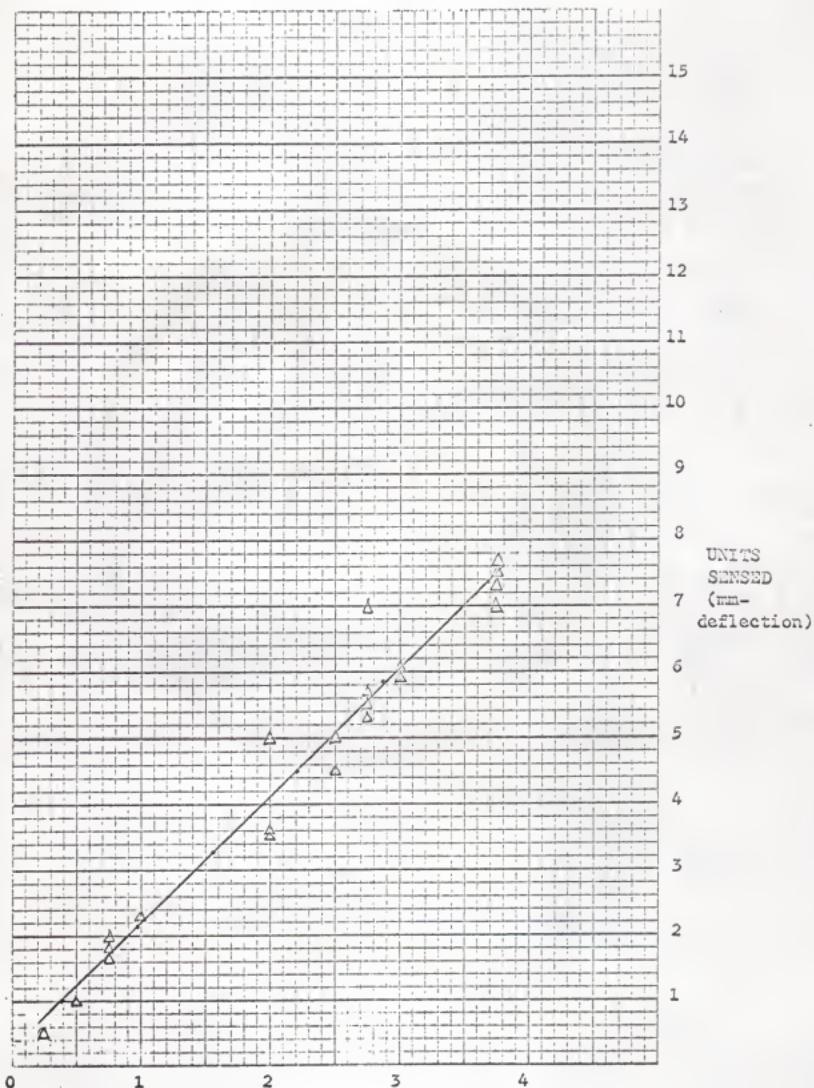
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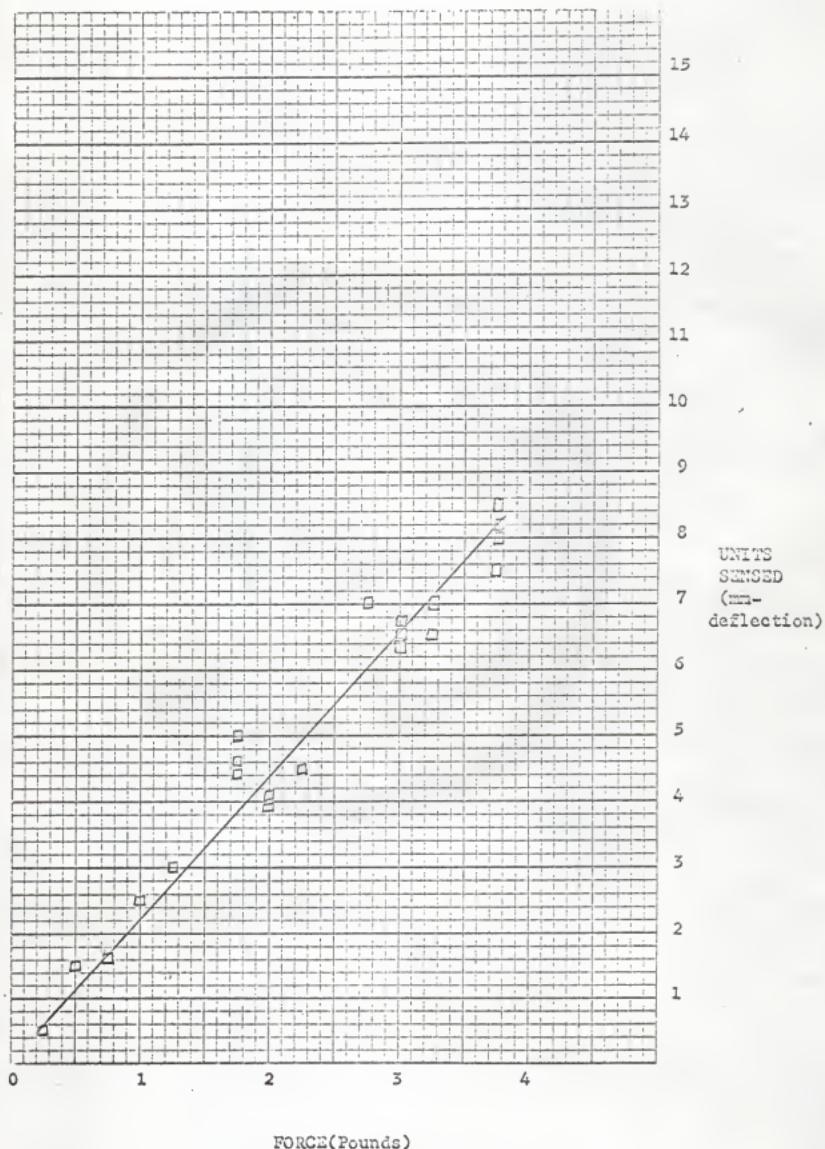
AXIS: Lateral DEADWEIGHT: 0

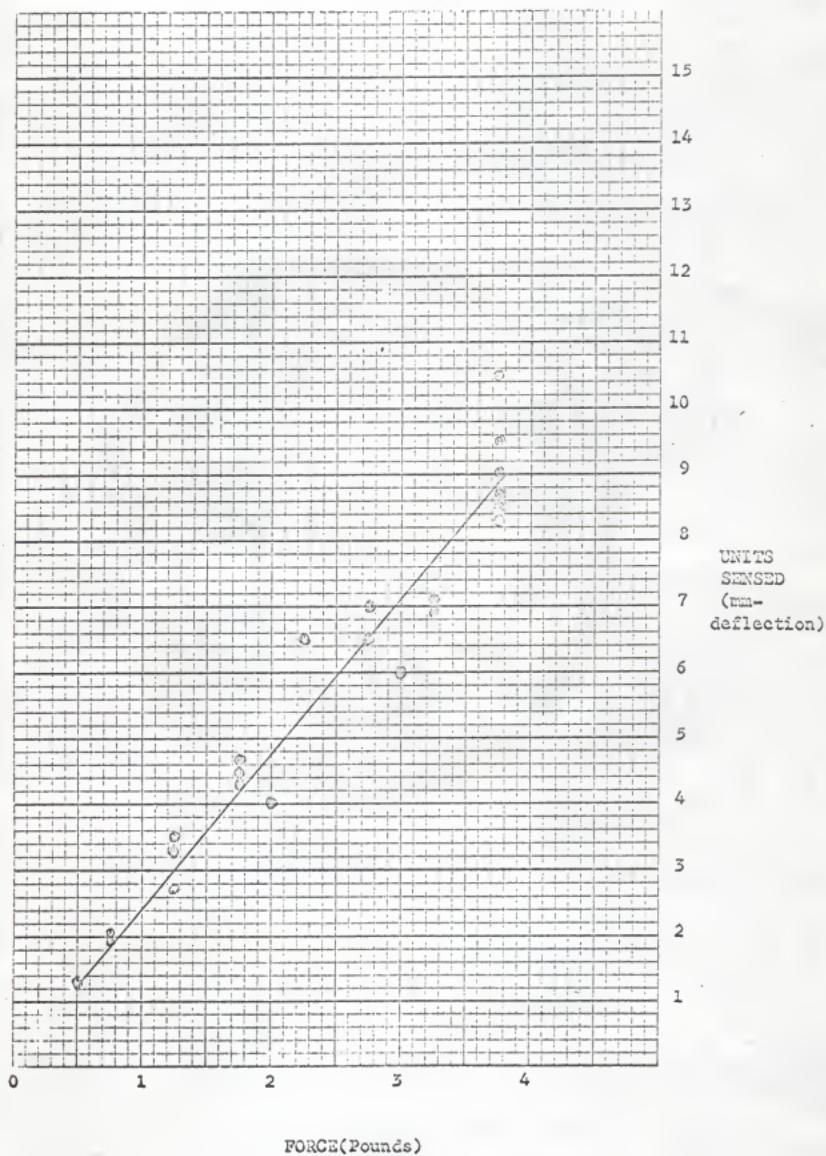


## STATIC CALIBRATION CURVES

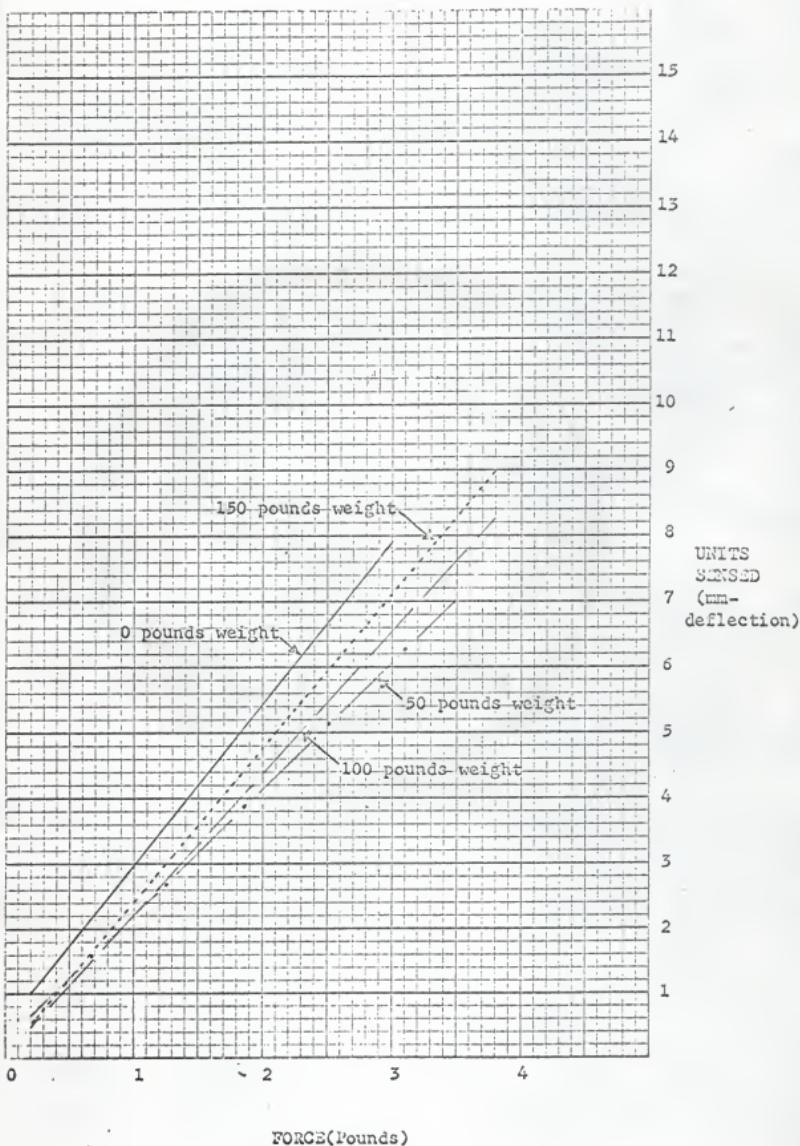
AXIS: Lateral DEADWEIGHT: 50

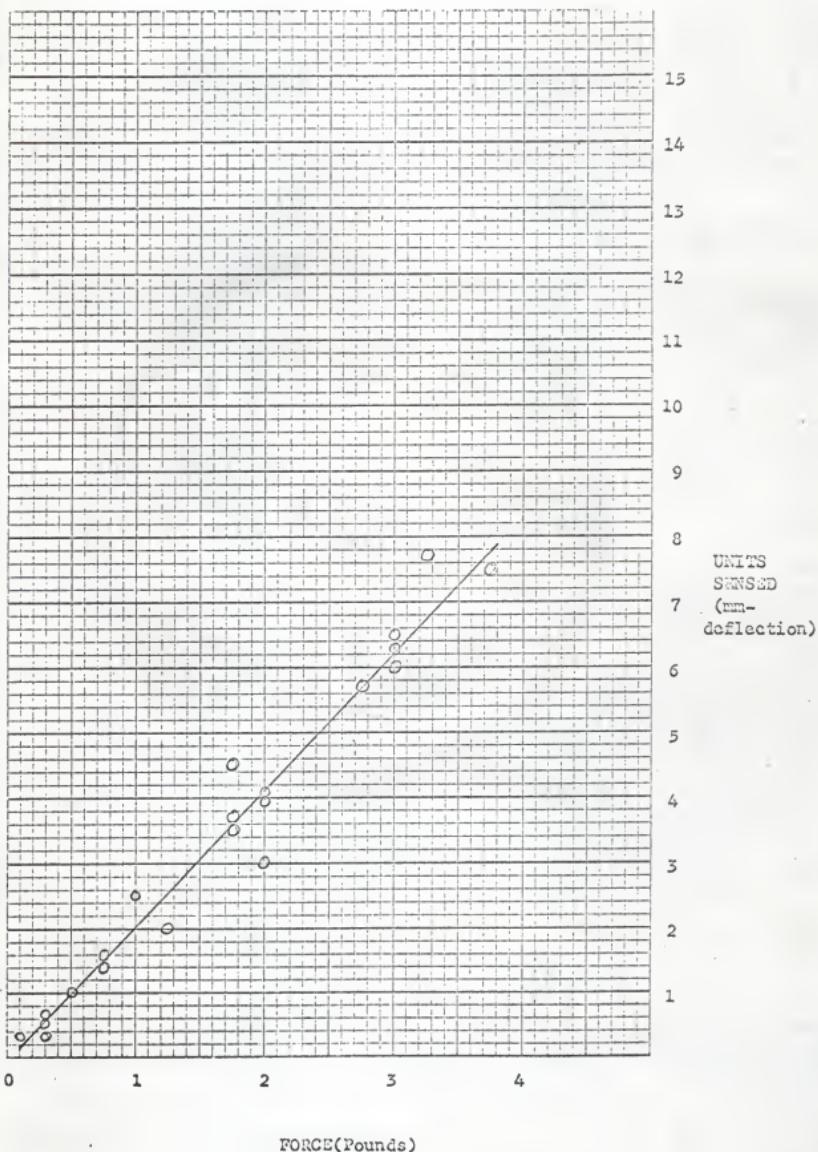


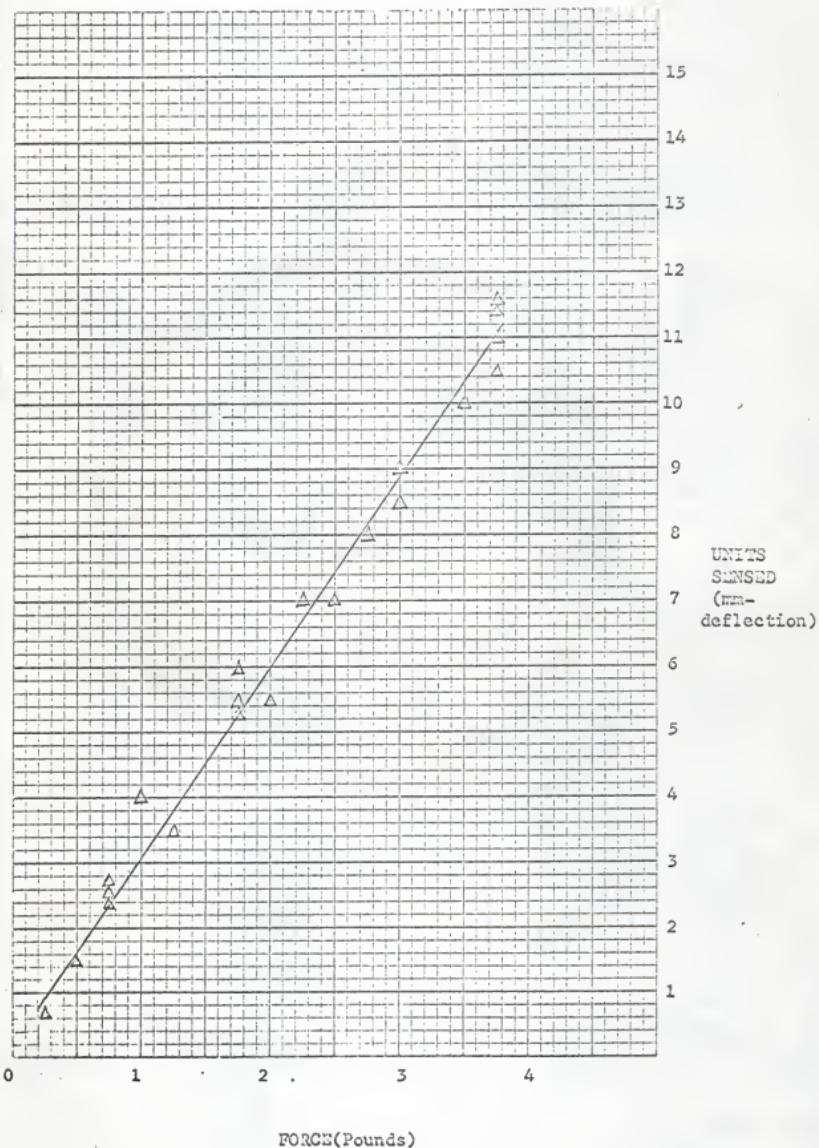
STATIC CALIBRATION CURVES  
AXIS: Lateral DEADWEIGHT: 100

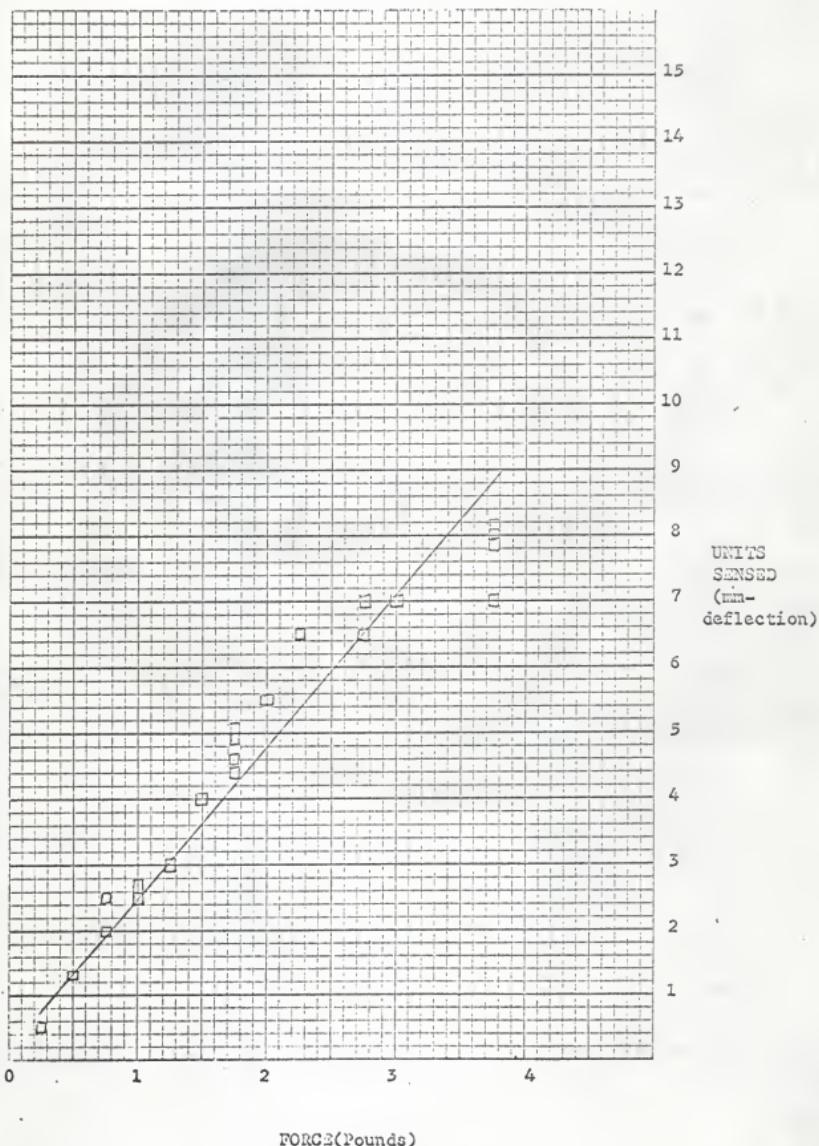
STATIC CALIBRATION CURVES  
AXIS: Lateral DEADWEIGHT: 150

STATIC CALIBRATION CURVES  
AXIS: Lateral DEADWEIGHT: 0,50,100,150

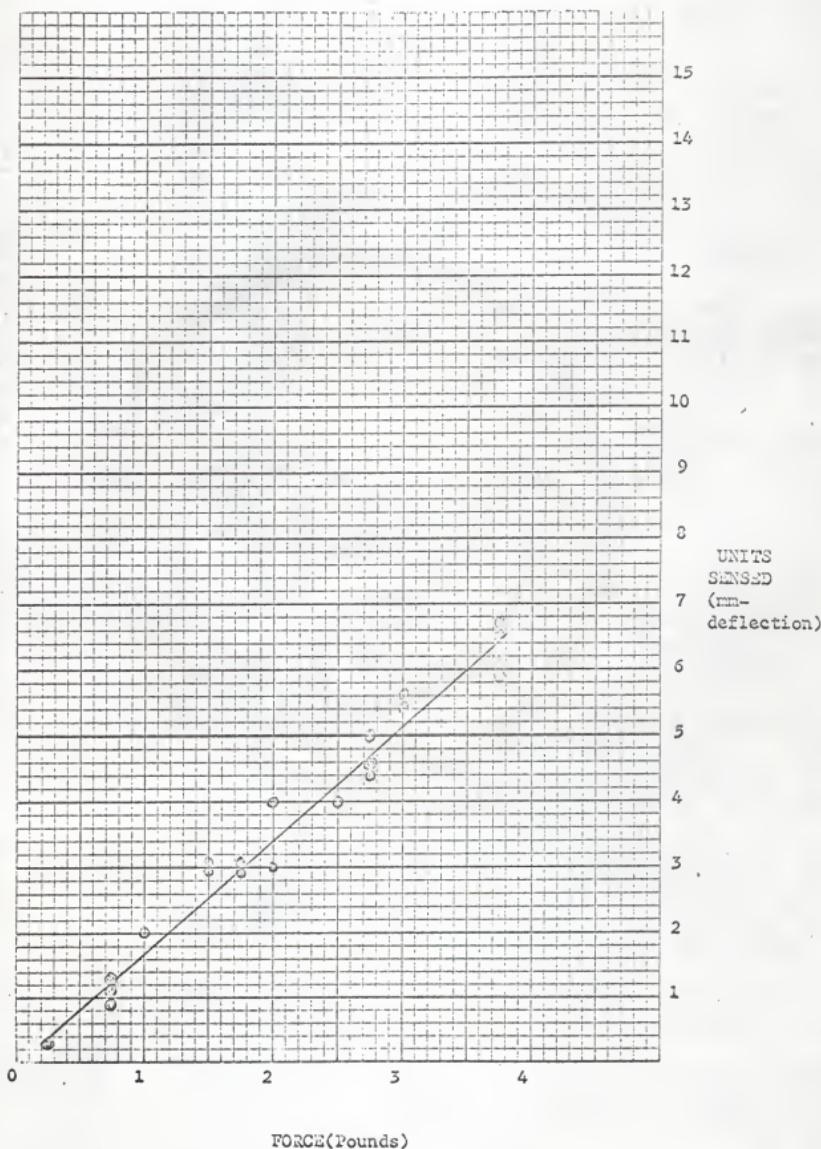


STATIC CALIBRATION CURVES  
AXIS: Frontal DEADWEIGHT: 0

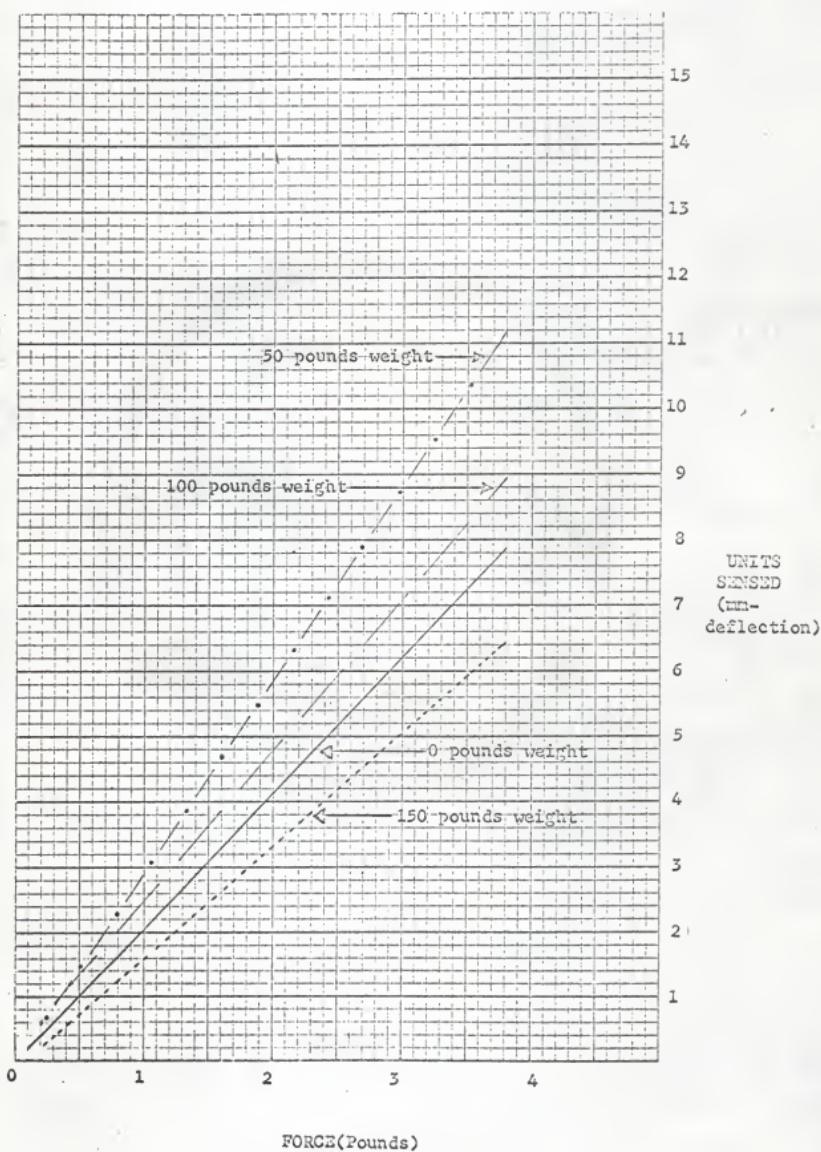
STATIC CALIBRATION CURVES  
AXIS: Frontal DEADWEIGHT: 50

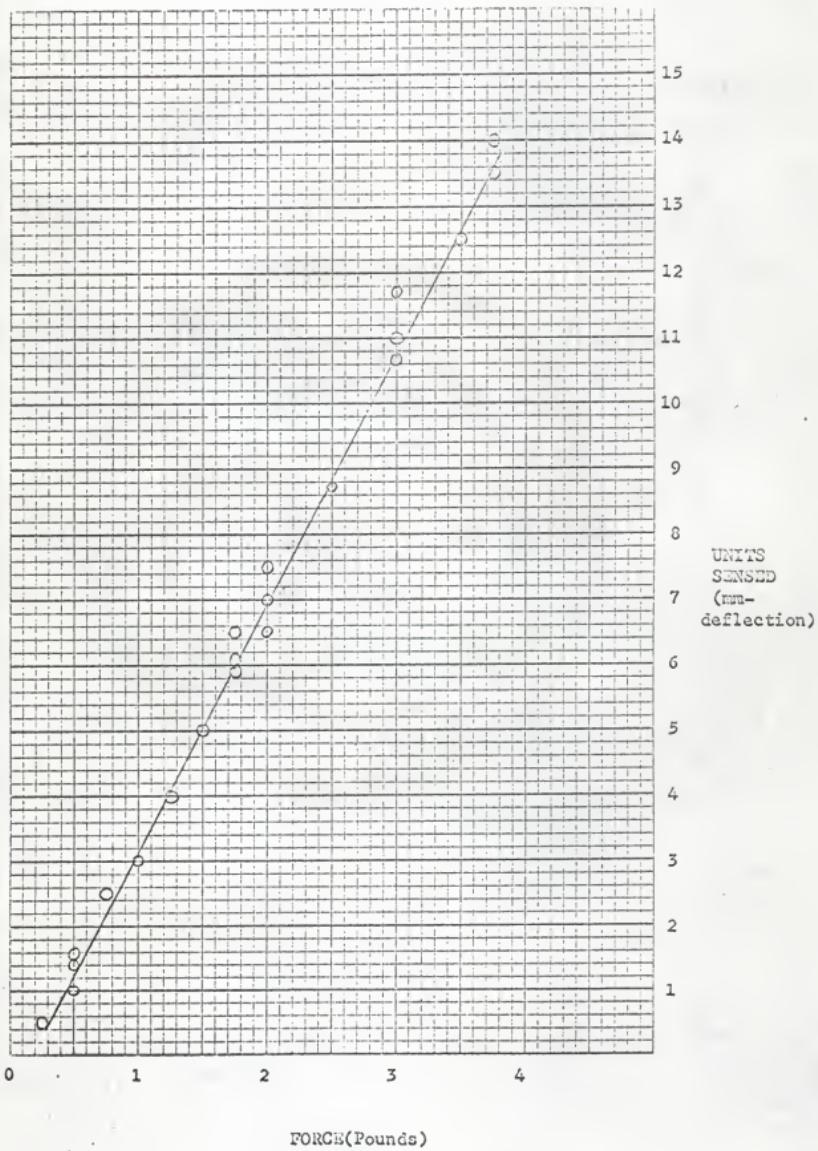
STATIC CALIBRATION CURVES  
AXIS: Frontal DEADWEIGHT: 100

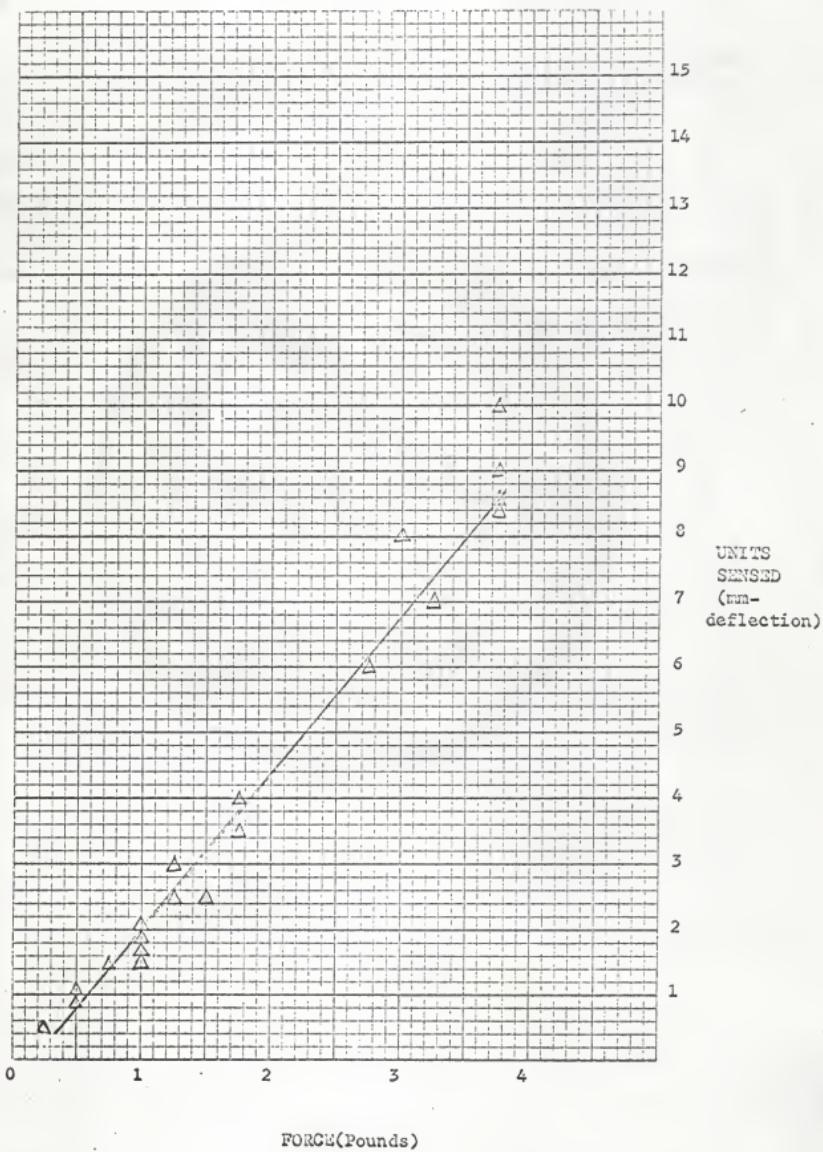
STATIC CALIBRATION CURVES  
AXIS: Frontal DEADWEIGHT: 150



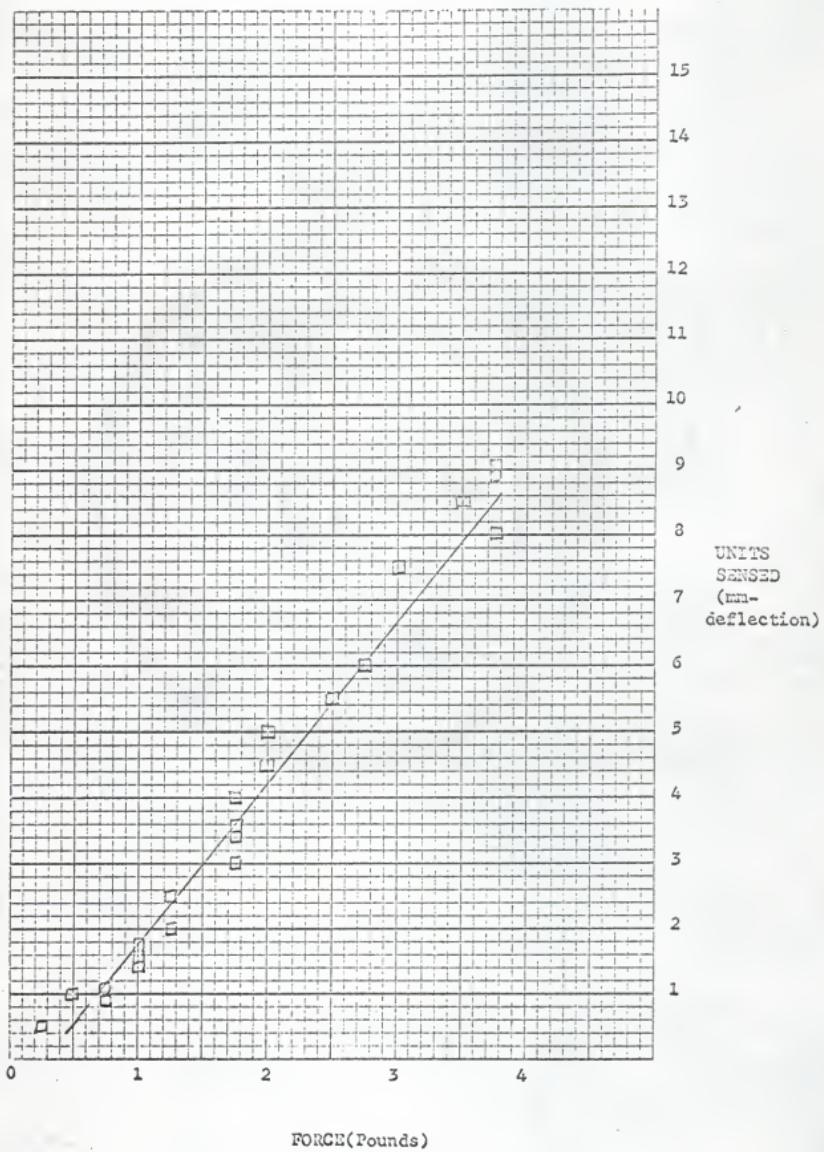
STATIC CALIBRATION CURVES  
AXIS: Frontal DEADWEIGHT: 0, 50, 100, 150



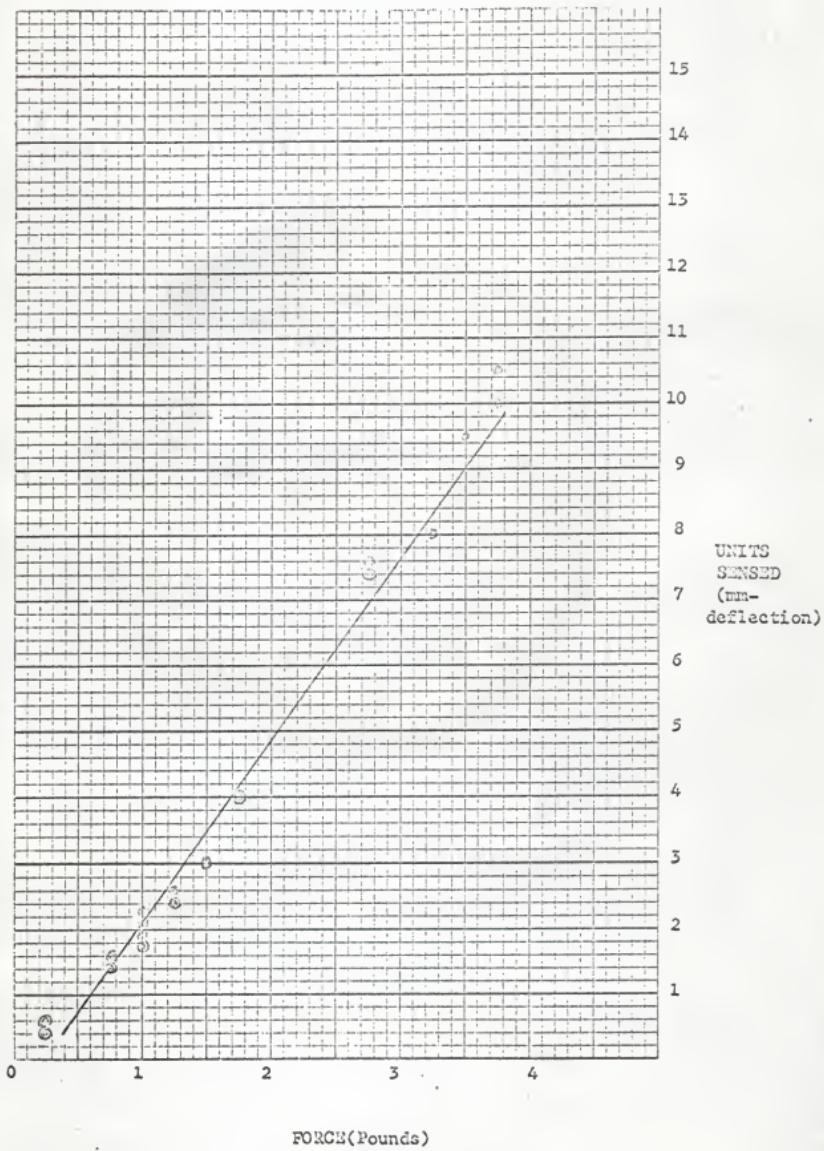
STATIC CALIBRATION CURVES  
AXIS: Vertical DEADWEIGHT: 0

STATIC CALIBRATION CURVES  
AXIS: Vertical DEADWEIGHT: 50

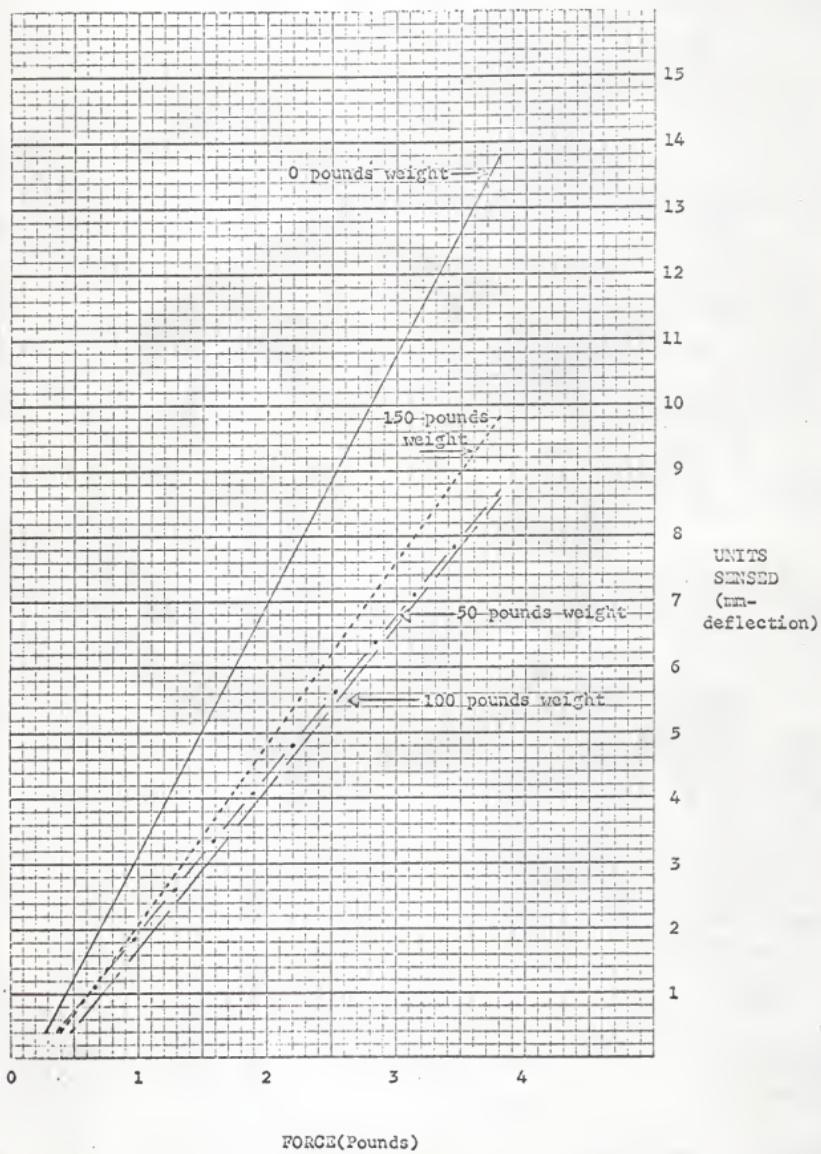
STATIC CALIBRATION CURVES  
AXIS: Vertical DEADWEIGHT: 100

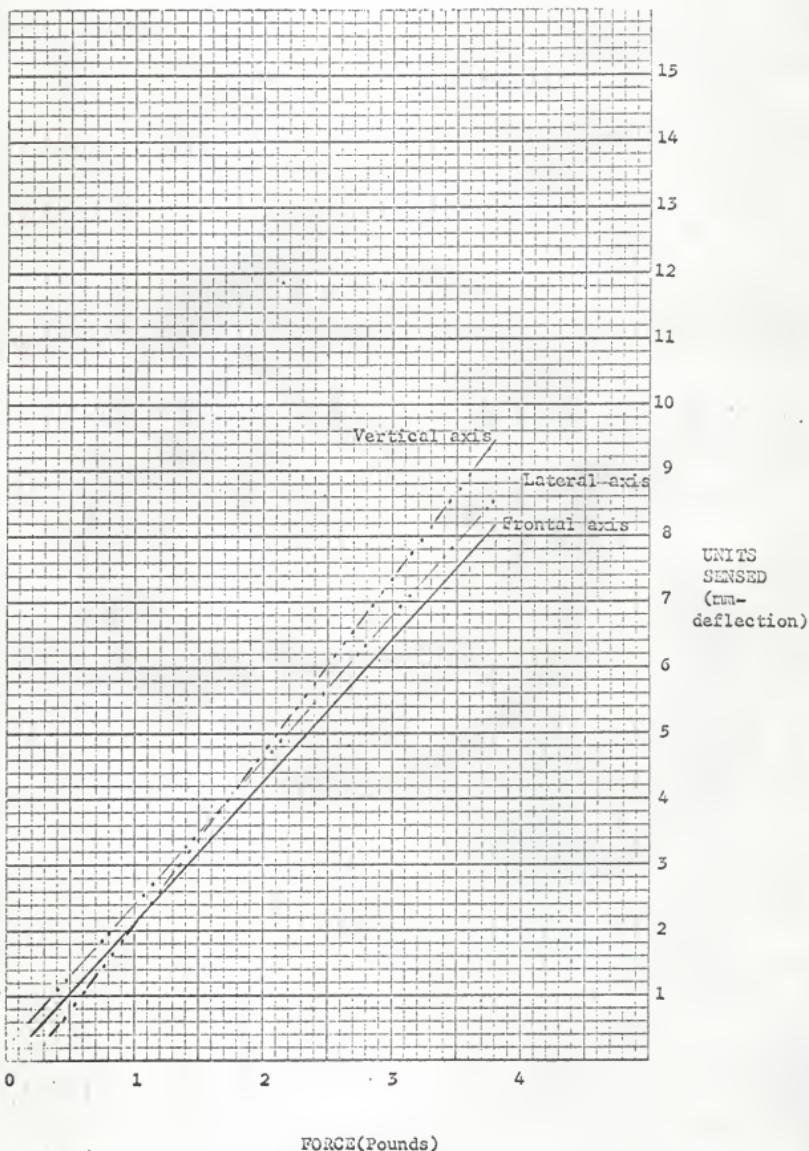


STATIC CALIBRATION CURVES  
AXIS: Vertical DEADWEIGHT: 150



STATIC CALIBRATION CURVES  
AXIS: Vertical DEFLECTION: 0, 50, 100, 150

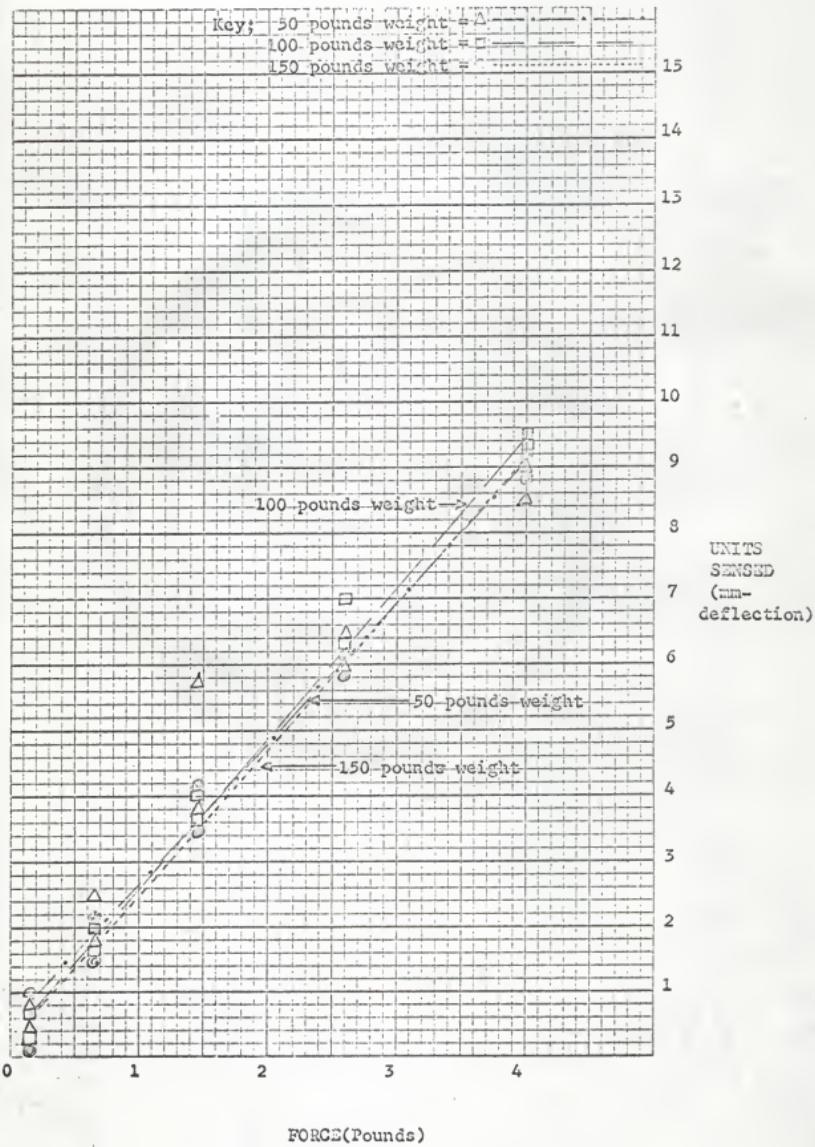


STATIC CALIBRATION CURVES  
AXES: All (Averages)

APPENDIX V

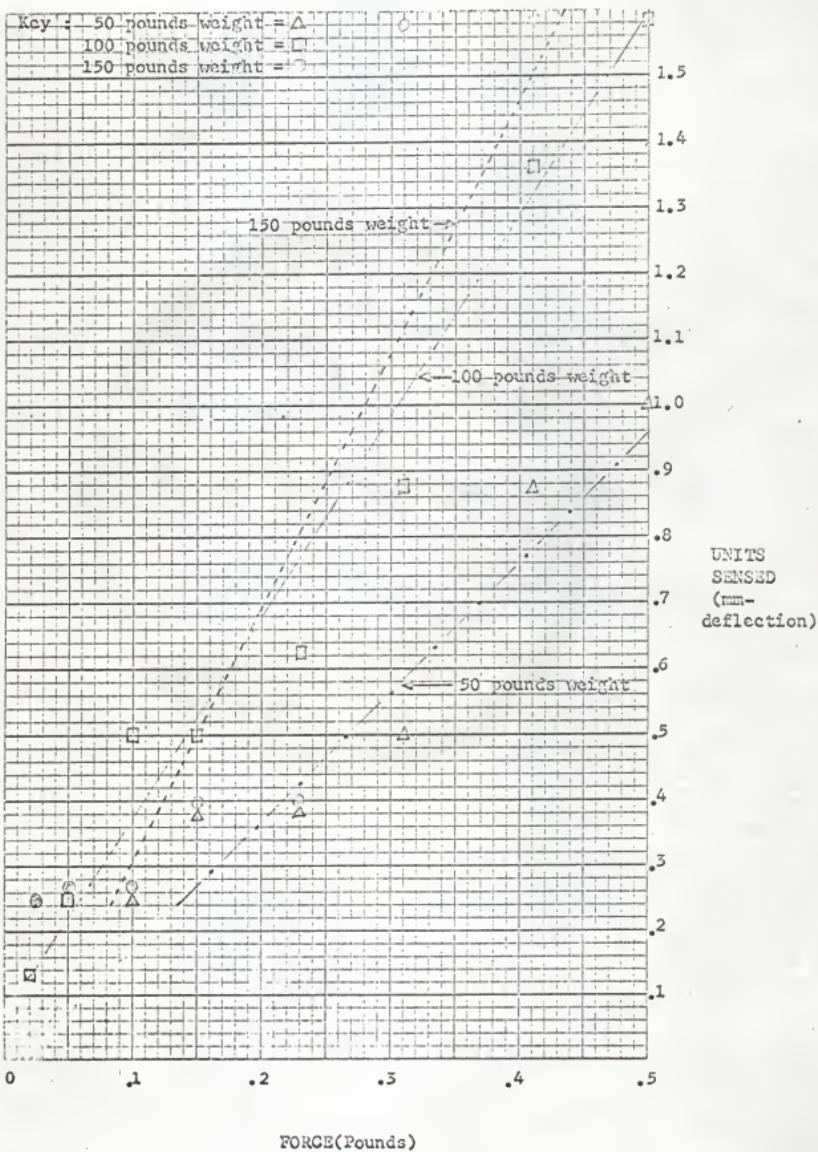
## DYNAMIC CALIBRATION CURVES

AXIS: Lateral DEADWEIGHT: 50,100,150  
(.25 pound revolved weight)



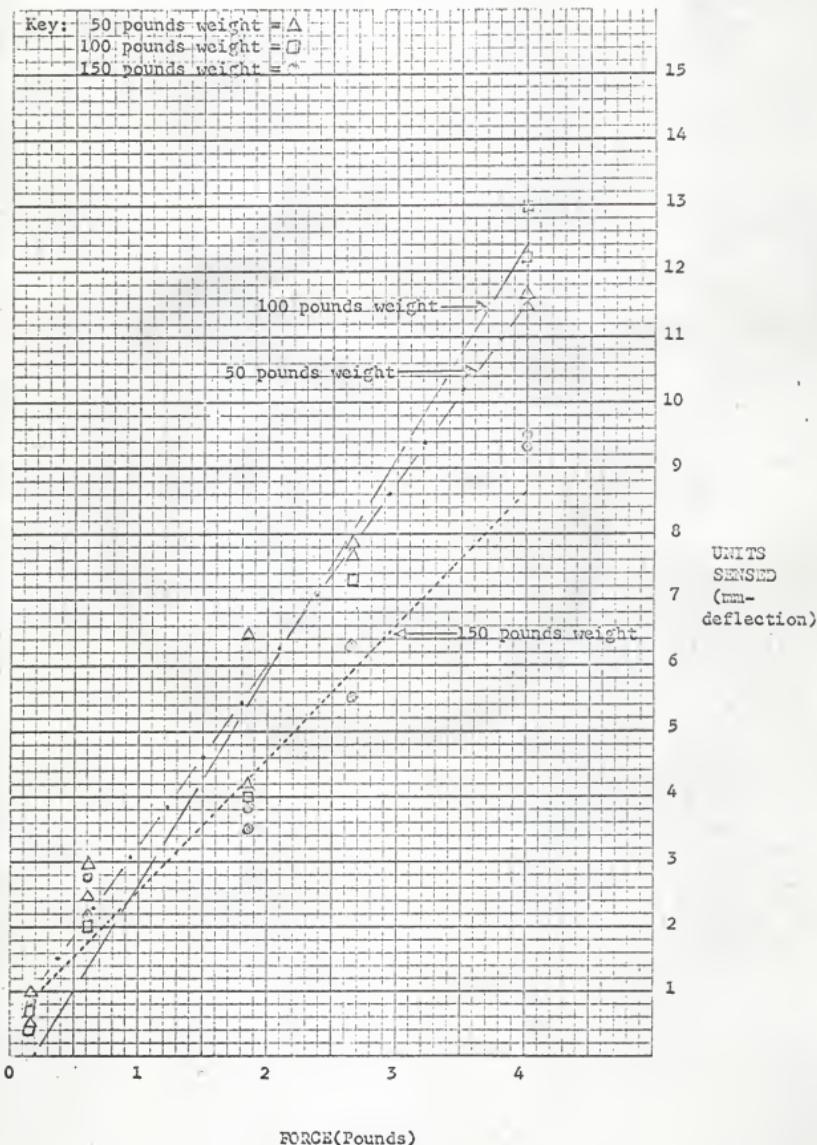
## DYNAMIC CALIBRATION CURVES

AXIS: Lateral DEAD WEIGHT: 50, 100, 150  
 (.01 pound weight-revolved)



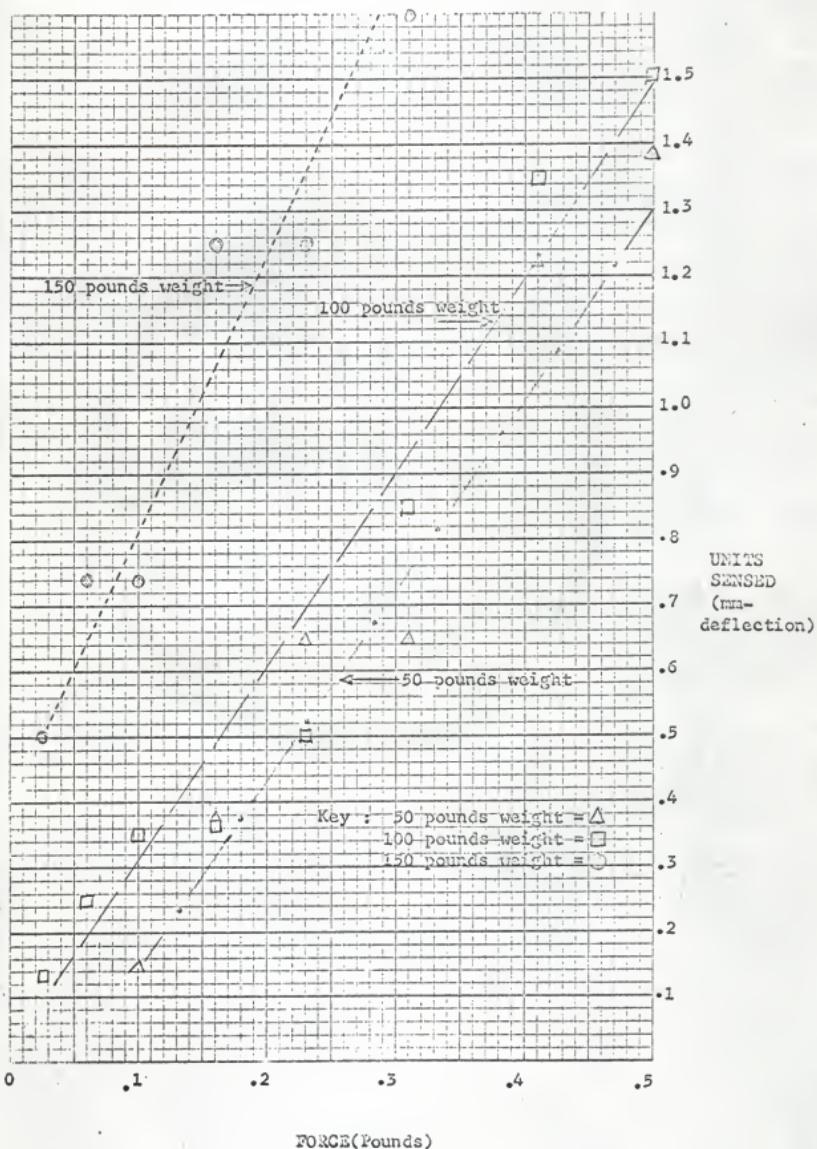
## DYNAMIC CALIBRATION CURVES

AXIS: Frontal DEADWEIGHT: 50,100,150  
(.25 pound revolved weight)



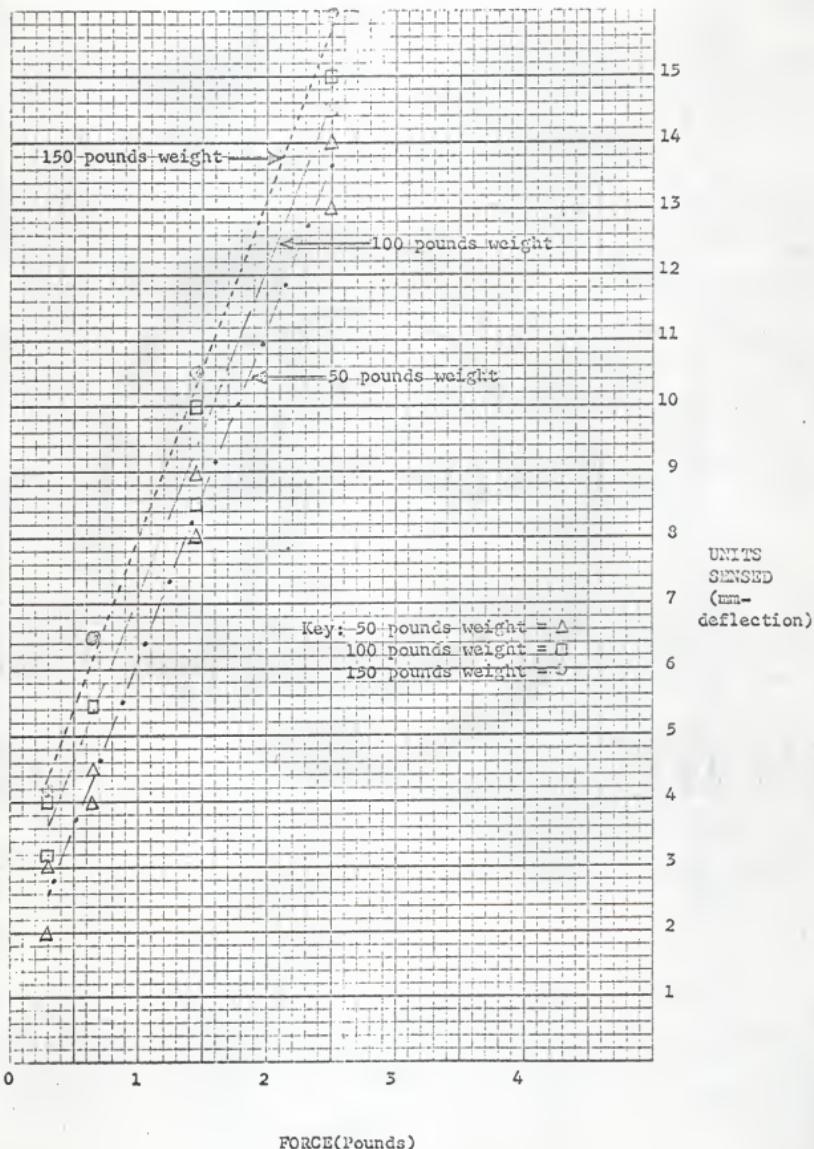
## DYNAMIC CALIBRATION CURVES

AXIS: Frontal DEADWEIGHT: 50,100,150  
 (.01 pound revolved weight)



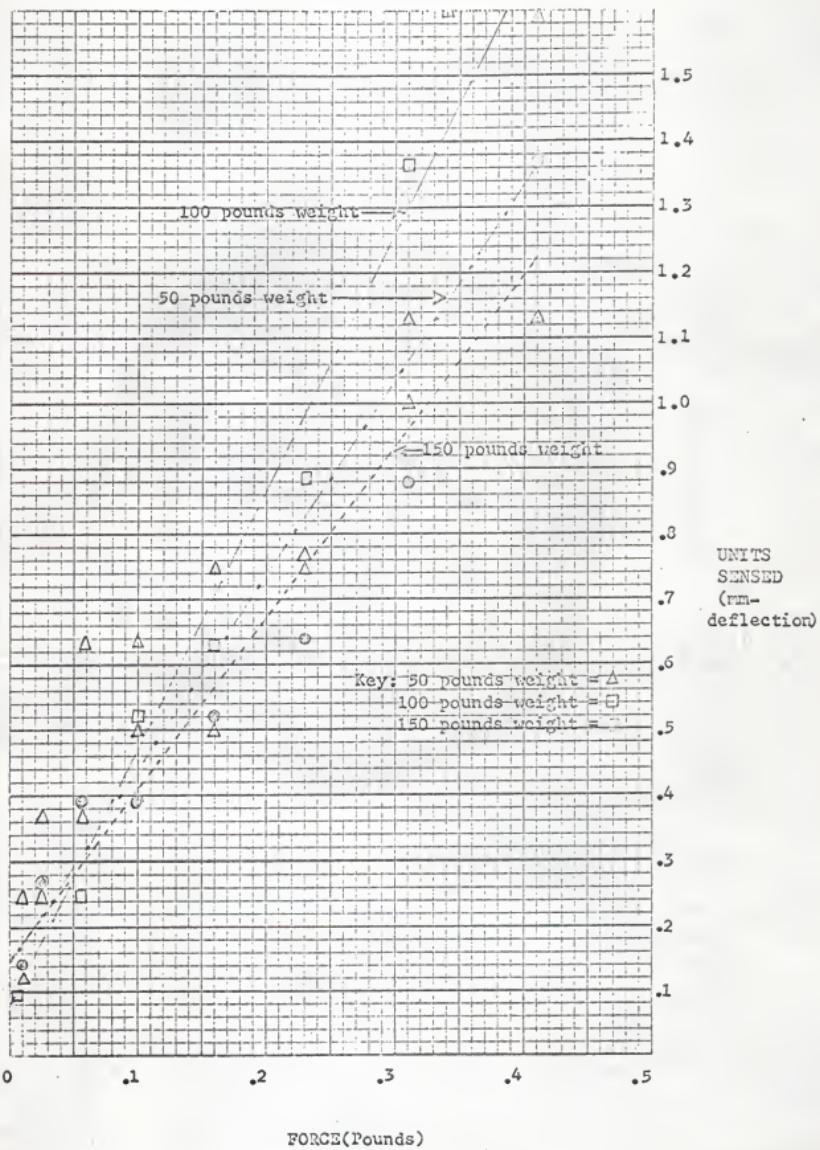
## DYNAMIC CALIBRATION CURVES

AXIS: Vertical DEADWEIGHT: .50, 100, 150  
(.25 pound revolved weight)

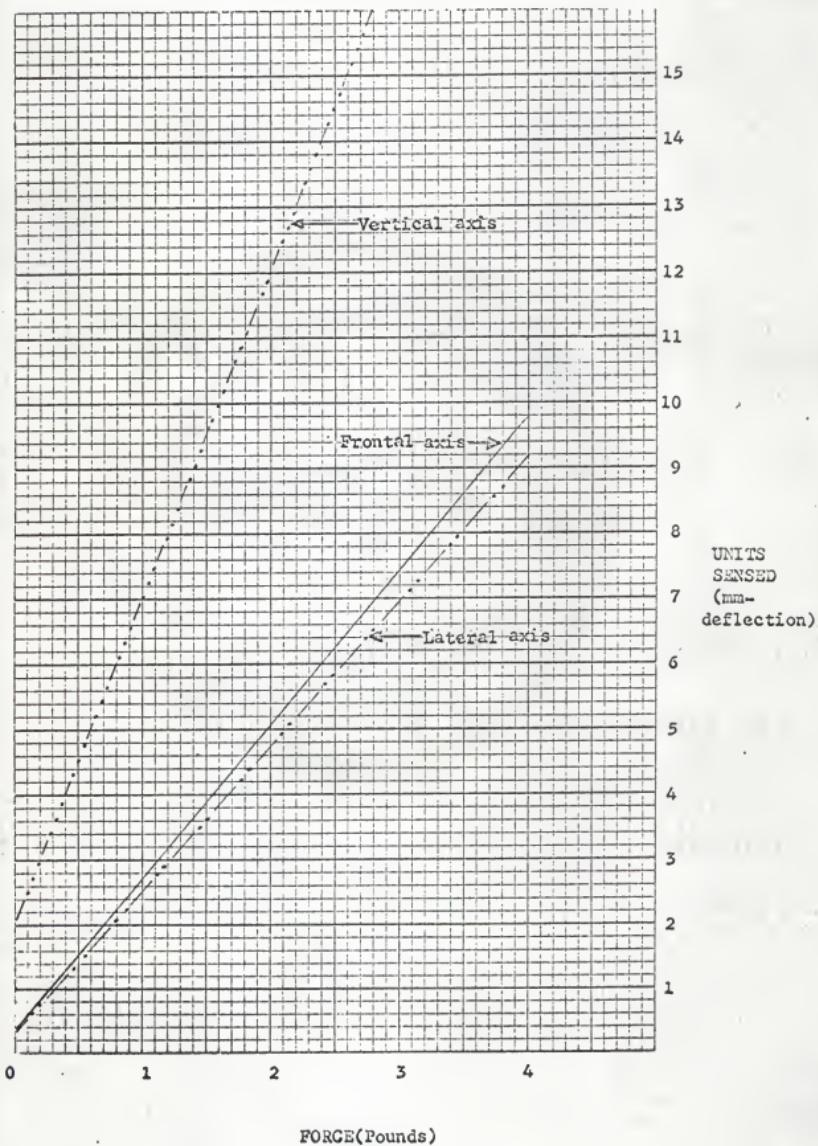


## DYNAMIC CALIBRATION CURVES

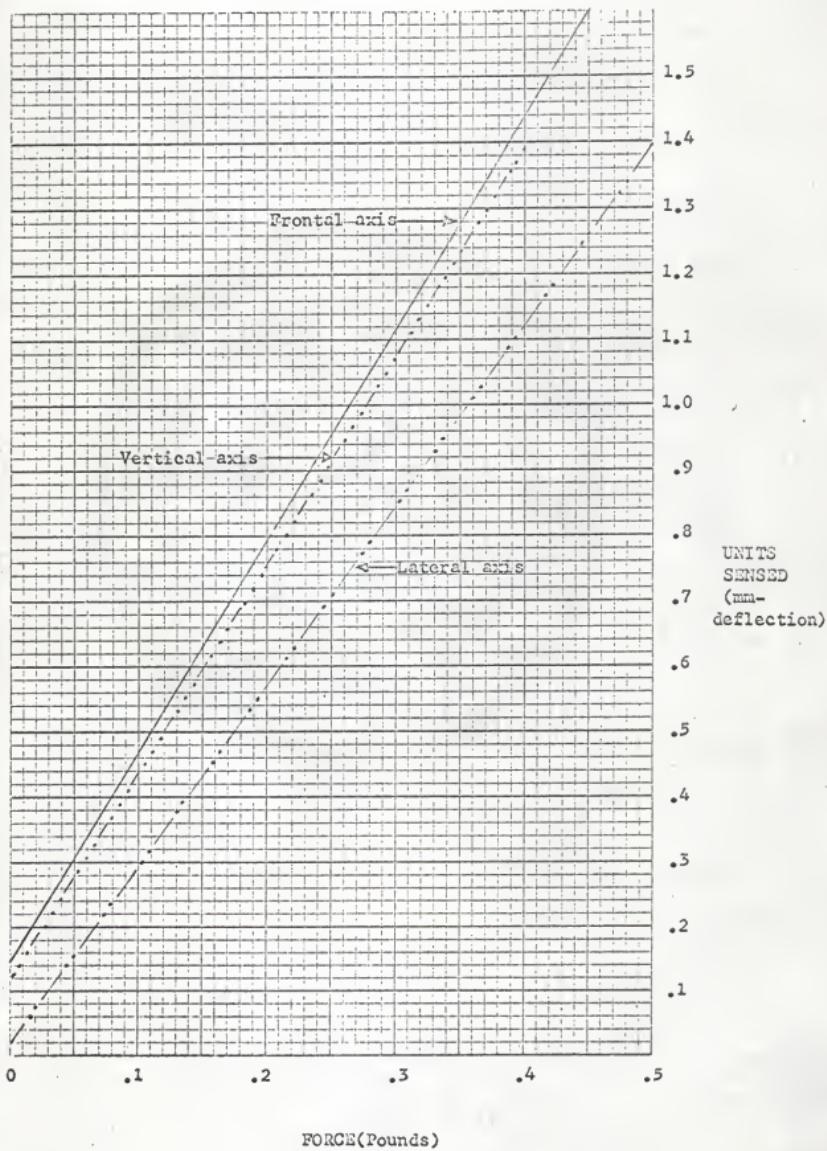
AXIS: Vertical DEADWEIGHT: 50,100,150  
(.01 pound revolved weight)



DYNAMIC CALIBRATION CURVES  
AXIS: All DEADWEIGHT: 50, 100, 150  
(.25 pound revolved weight)

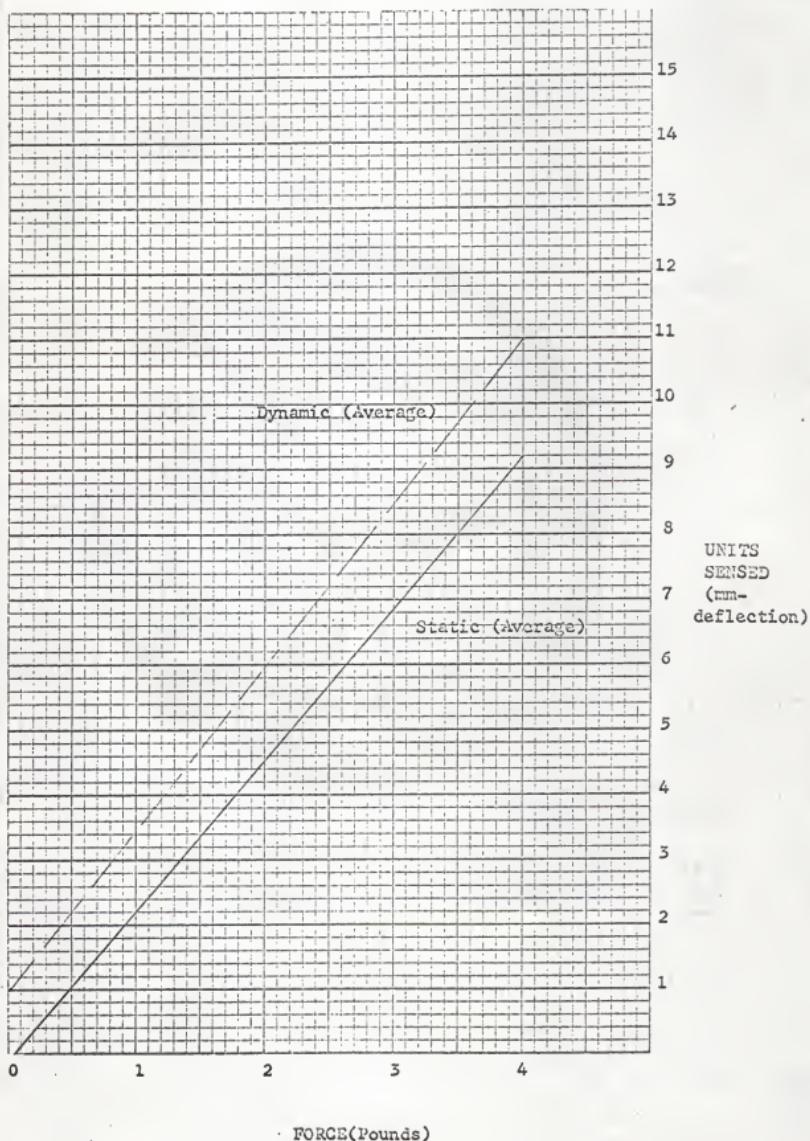


DYNAMIC CALIBRATION CURVES  
AXIS: All DIA-MON-SIGHT: 50,100,150  
(.01 pound revolved weight)



APPENDIX VI

## STATIC AND DYNAMIC CALIBRATION CURVES



DESIGN AND CONSTRUCTION OF AN  
INSTRUMENT TO DYNAMICALLY EVALUATE  
A FORCE PLATFORM

by

NATHAN DOYLE GRANGER

B. S., Kansas State University, 1966

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AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Industrial Engineering

KANSAS STATE UNIVERSITY  
Manhattan, Kansas

1967

## ABSTRACT

The performance characteristics of Hearn's force platform were investigated.

A dynamic calibration instrument which consisted of a rheostat controlled, AC-DC motor, which revolved a known weight about its shaft was designed and constructed.

This instrument developed a centripetal force which was recorded as a sine wave on a strip chart recorder. By measuring the amplitude of this sine wave and calculating the corresponding centripetal force, an input-output curve could be recorded. Analysis of these graphs gave the following results:

- (1) The sensitivity of the present platform was determined to be .01 pound force in the vertical axis and .02 and .025 in the lateral and frontal axes, respectively. A heartbeat was recorded as a .5 pound force in the frontal axis.
- (2) Calibration was found to be unaffected by loading the platform with a deadweight of 0, 50, 100, or 150 pounds.
- (3) The operating frequency of the platform was tested between 30 and 540 CPM; it was satisfactory within this range. A natural frequency (maximum) was determined to be 3500 CPM.
- (4) The platform will respond similarly to either dynamic or static forces. These responses approach equality when induced under similar experimental conditions.
- (5) The dynamic calibration technique for the vertical axis is not valid.